TURBOCHARGER

TURBO THEORY

A turbo charger is basically an exhaust gas driven air compressor and can be best understood if it is divided into its two basic parts, the exhaust gas driven turbine and its housing, and the air compressor and its housing. I did say divided didn't I. Well I should have said like a set of Siamese twins because each of them performs different functions but, because they are joined together at the hip via a common shaft, the function of one impacts the function of the other. How? Take a perfectly set up compressor section and mate it with an incorrect turbine section, or visa versa, and you end up with our Siamese twins trying to go in different directions. The result is that our Siamese twins end up wasting all of their energy fighting each other and go nowhere.

When considering a turbo charger most folks tend to look at the maximum CFM rating of the compressor and ignore everything else under the assumption that the compressor and the exhaust turbine are perfectly matched out of the box. I will grant you that in stock factory applications that are probably close to the truth but, in all out performance applications, nothing could be further from the truth because of the extremes of operation in a performance application.

The goal in a performance application is to get the exhaust turbine up to speed as quickly as possible however; it must be mated to a compressor wheel that will generate as much pressure as it can as soon as possible. This is a contradiction because the exhaust turbine generates the drive power and the compressor consumes that power. The larger the compressor and the higher the pressure (boost) we want, the quicker the power from the exhaust turbine is used up. Put in a larger exhaust turbine and it will take the engine longer to develop enough hot expanding exhaust gas to spin it, slowing down the compressor and causing turbo lag. At this point I am going to repeat something stated earlier, do not think of a turbo charger as a bolt on piece of equipment, think of it as a system.

The turbine is powered by hot expanding exhaust gas, a lot of hot expanding exhaust gas, the more and the hotter the expanding exhaust gas the better. I am sure many of you have seen pictures of turbo charged engines with cherry red hot exhaust systems and turbo housings. The captions under most of these types of pictures proclaim outstanding horse power numbers. What most of the articles related to these pictures do not tell you is that the engine was under an extreme load. A load so heavy that the engine was almost at its stall point for a prolonged period of time. A condition that most turbo charged engines will never see.

The real point I am trying to make is that the exhaust turbine will not generate enough power to turn the air compressor fast enough for it to work properly unless the engine is

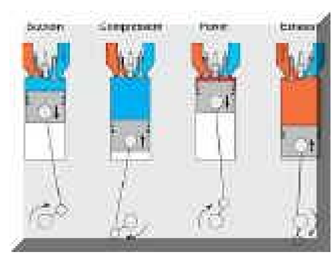
feeding the exhaust turbine a lot of hot expanding exhaust gas, a condition that can only be created when the engine is under a load. There is where the selection of transmission gear ratios and the ring and pinion ratio play a critical part. The fact that the engine must be under a load is the reason why, no matter how high you rev a turbo charged engine with no load on it, you will not see the boost gauge move.

This is also where the term 'turbo lag' came from. Turbo lag is basically the amount of time it takes from the time you place a load on the engine (stomp the gas peddle to the floor and dump the clutch or, get full converter lock up with your automatic trans) until the time the engine develops enough hot expanding exhaust gas to spin the turbine fast enough for the compressor to do its job.

Effectively, a turbo charged engine is a normally aspirated engine until the turbine and compressor spin up. To minimize turbo lag, it is imperative that the turbine and the compressor are properly matched to the engine as well as the engine being properly matched to the transmission gears, the ring and pinion gears, and the tires.

TURBO PRINCIPLES

To better understand the technique of turbocharging, it is useful to be familiar with the internal combustion engine's principles of operation. Today, most passenger car and commercial diesel engines are four-stroke piston engines controlled by intake and exhaust valves. One operating cycle consists of four strokes during two complete revolutions of the crankshaft.



Suction (charge exchange stroke)
When the piston moves down, air (diesel engine or direct injection petrol engine) or a fuel/air mixture (petrol engine) is drawn through the intake valve.

Compression (power stroke)
The cylinder volume is compressed.

Expansion (power stroke)
In the petrol engine, the fuel/air mixture is ignited by a spark plug, whereas in the diesel engine fuel is injected under high pressure and the mixture ignites spontaneously.

Exhaust (charge exchange stroke)
The exhaust gas is expelled when the piston moves up.

These simple operating principles provide various possibilities of increasing the engine's power output:

Swept volume enlargement

Enlargement of the swept volume allows for an increase in power output, as more air is available in a larger combustion chamber and thus more fuel can be burnt. This enlargement can be achieved by increasing either the number of cylinders or the volume of each individual cylinder. In general, this results in larger and heavier engines. As far as fuel consumption and emissions are concerned, no significant advantages can be expected.

Increase in engine rpm

Another possibility for increasing the engine's power output is to increase its speed. This is done by increasing the number of firing strokes per time unit. Because of mechanical stability limits, however, this kind of output improvement is limited. Furthermore, the increasing speed makes the frictional and pumping losses increase exponentially and the engine efficiency drops.

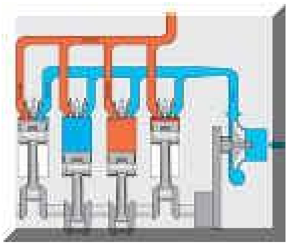
Turbocharging

In the above-described procedures, the engine operates as a naturally aspirated engine. The combustion air is drawn directly into the cylinder during the intake stroke. In turbocharged engines, the combustion air is already pre-compressed before being supplied to the engine. The engine aspirates the same volume of air, but due to the higher pressure, more air mass is supplied into the combustion chamber. Consequently, more fuel can be burnt, so that the engine's power output increases related to the same speed and swept volume.

Basically, one must distinguish between mechanically supercharged and exhaust gas turbocharged engines.

Mechanical supercharging

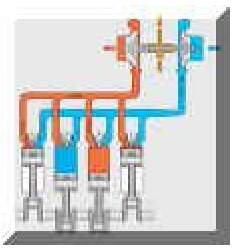
With mechanical supercharging, the combustion air is compressed by a compressor driven directly by the engine. However, the power output increase is partly lost due to the parasitic losses from driving the compressor. The power to drive a mechanical turbocharger is up to 15 & percent; of the engine output. Therefore, fuel consumption is higher when compared with a naturally aspirated engine with the same power output.



Schematic of a mechanically supercharged four-cylinder engine

Exhaust gas turbocharging

In exhaust gas turbocharging, some of the exhaust gas energy, which would normally be wasted, is used to drive a turbine. Mounted on the same shaft as the turbine is a compressor which draws in the combustion air, compresses it, and then supplies it to the engine. There is no mechanical coupling to the engine.

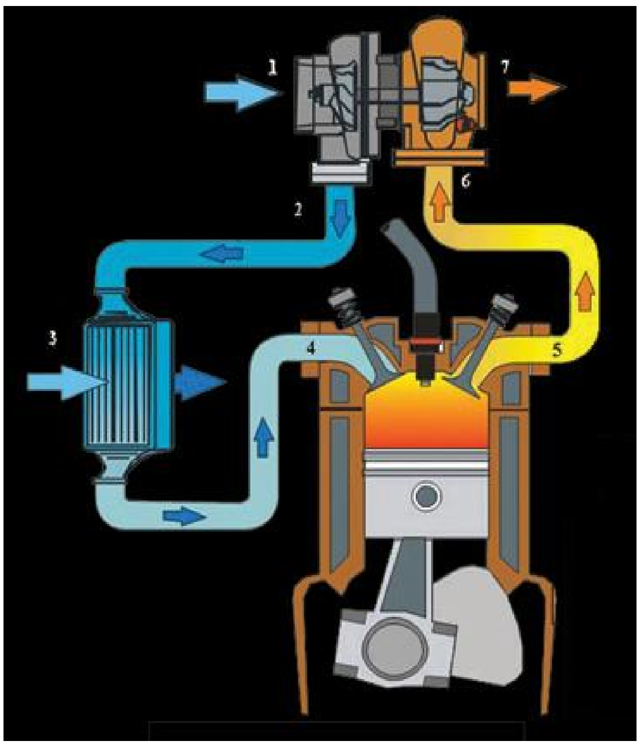


Schematic of an exhaust gas turbocharged four-cylinder

How a Turbo System Works

Engine power is proportional to the amount of air and fuel that can get into the cylinders. All things being equal, larger engines flow more air and as such will produce more power. If we want our small engine to perform like a big engine, or simply make our bigger engine produce more power, our ultimate objective is to draw more air into the cylinder. By installing a Garrett turbocharger, the power and performance of an engine can be dramatically increased.

So how does a turbocharger get more air into the engine? Let us first look at the schematic below:



1 Compressor Inlet
2 Compressor Discharge
3 Charge air cooler (CAC)
4 Intake Valve
5 Exhaust Valve
6 Turbine Inlet
7 Turbine Discharge

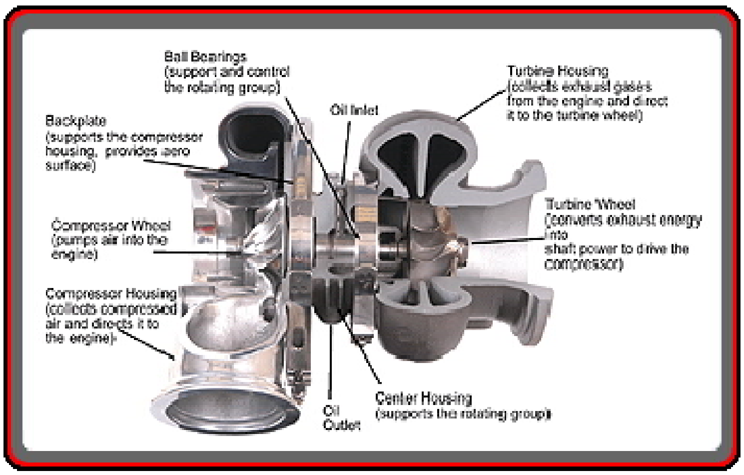
The components that make up a typical turbocharger system are:

- The air filter (not shown) through which ambient air passes before entering the compressor (1)
- The air is then compressed which raises the air's density (mass / unit volume) (2)
- Many turbocharged engines have a charge air cooler (aka intercooler) (3) that cools the compressed air to further increase its density and to increase resistance to detonation
- After passing through the intake manifold (4), the air enters the engine's cylinders, which contain a fixed volume. Since the air is at elevated density, each

cylinder can draw in an increased mass flow rate of air. Higher air mass flow rate allows a higher fuel flow rate (with similar air/fuel ratio). Combusting more fuel results in more power being produced for a given size or displacement

- After the fuel is burned in the cylinder it is exhausted during the cylinder's exhaust stroke in to the exhaust manifold (5)
- The high temperature gas then continues on to the turbine (6). The turbine creates backpressure on the engine which means engine exhaust pressure is higher than atmospheric pressure
- A pressure and temperature drop occurs (expansion) across the turbine (7), which harnesses the exhaust gas' energy to provide the power necessary to drive the compressor

What are the components of a turbocharger?



The turbocharger has four main components. The turbine (almost always a radial turbine) and impeller/compressor wheels are each contained within their own folded conical housing on opposite sides of the third component, the center housing/hub rotating assembly (CHRA).

The housings fitted around the compressor impeller and turbine collect and direct the gas flow through the wheels as they spin. The size and shape can dictate some performance characteristics of the overall turbocharger. Often the same basic turbocharger assembly will be available from the manufacturer with multiple housing choices for the turbine and sometimes the compressor cover as well. This allows the designer of the engine system to tailor the compromises between performance, response, and efficiency to application or preference.

The turbine and impeller wheel sizes also dictate the amount of air or exhaust that can be flowed through the system, and the relative efficiency at which they operate. Generally, the larger the turbine wheel and compressor wheel, the larger the flow capacity. Measurements and shapes can vary, as well as curvature and number of blades on the wheels.

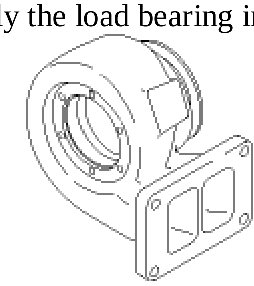
The center hub rotating assembly houses the shaft which connects the compressor impeller and turbine. It also must contain a bearing system to suspend the shaft, allowing it to rotate at very high speed with minimal friction. For instance, in automotive applications the CHRA typically uses a thrust bearing or ball bearing lubricated by a constant supply of pressurized engine oil. The CHRA may also be considered "water cooled" by having an entry and exit point for engine coolant to be cycled. Water cooled models allow engine coolant to be

The layout of the turbocharger in a given application is critical to a properly performing system. Intake and exhaust plumbing is often driven primarily by packaging constraints. We will explore exhaust manifolds in more detail in subsequent tutorials; however, it is important to understand the need for a compressor bypass valve (commonly referred to as a Blow-Off valve) on the intake tract and a Wastegates for the exhaust flow.

Turbine Housing

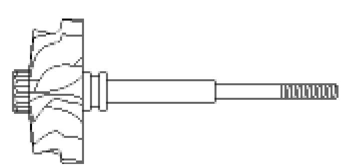
Turbine housings are manufactured in various grades of spheroidal graphite iron to deal with thermal fatigue and wheel burst containment. As with the impeller, profile machining to suit turbine blade shape is carefully controlled for optimum performance.

The turbine housing inlet flange acts as the reference point for fixing turbocharger position relative to its installation. It is normally the load bearing interface.



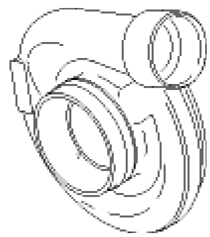
Wheel

The Turbine Wheel is housed in the turbine casing and is connected to a shaft that in turn rotates the compressor wheel.



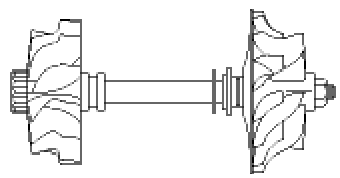
Compressor Cover

Compressor housings are also made in cast aluminum. Various grades are used to suit the application. Both gravity die and sand casting techniques are used. Profile machining to match the developed compressor blade shape is important to achieve performance consistency.



Compressor Wheel (Impellor)

Compressor impellers are produced using a variant of the aluminum investment casting process. A rubber former is made to replicate the impeller around which a casting mould is created. The rubber former can then be extracted from the mould into which the metal is poured. Accurate blade sections and profiles are important in achieving compressor performance. Back face profile machining optimizes impeller stress conditions. Boring to tight tolerance and burnishing assist balancing and fatigue resistance. The impeller is located on the shaft assembly using a threaded nut.



Other Components

Blow-Off (Bypass) Valves

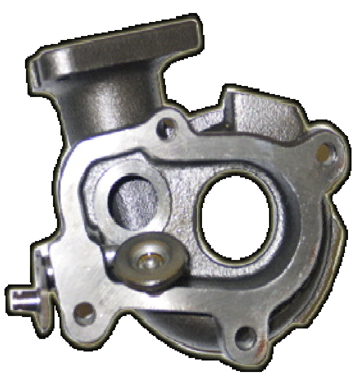
The Blow-Off valve (BOV) is a pressure relief device on the intake tract to prevent the turbo’s compressor from going into surge. The BOV should be installed between the compressor discharge and the throttle body, preferably downstream of the charge air cooler (if equipped). When the throttle is closed rapidly, the airflow is quickly reduced, causing flow instability and pressure fluctuations. These rapidly cycling pressure fluctuations are the audible evidence of surge. Surge can eventually lead to thrust bearing failure due to the high loads associated with it.

Blow-Off valves use a combination of manifold pressure signal and spring force to detect when the throttle is closed. When the throttle is closed rapidly, the BOV vents boost in the intake tract to atmosphere to relieve the pressure; helping to eliminate the phenomenon of surge.

Wastegates
On the exhaust side, a Wastegates provides us a means to control the boost pressure of the engine. Some commercial diesel applications do not use a Wastegates at all. This type of system is called a free-floating turbocharger.

However, the vast majority of gasoline performance applications require a Wastegates. There are two (2) configurations of Wastegates, **internal** or **external**. Both internal and external Wastegates provide a means to bypass exhaust flow from the turbine wheel. Bypassing this energy (e.g. exhaust flow) reduces the power driving the turbine wheel to match the power required for a given boost level. Similar to the BOV, the Wastegates uses boost pressure and spring force to regulate the flow bypassing the turbine.

Internal Wastegates are built into the turbine housing and consist of a “flapper” valve, crank arm, rod end, and pneumatic actuator. It is important to connect this actuator only to boost pressure; i.e. it is not designed to handle vacuum and as such should not be referenced to an intake manifold.



External Wastegates are added to the exhaust plumbing on the exhaust manifold or header. The advantage of external Wastegates is that the bypassed flow can be reintroduced into the exhaust stream further downstream of the turbine. This tends to improve the turbine’s performance. On racing applications, this Wastegated exhaust flow can be vented directly to atmosphere.

Oil & Water Plumbing

The intake and exhaust plumbing often receives the focus leaving the oil and water plumbing neglected.

Garrett ball bearing turbochargers require less oil than journal bearing turbos. Therefore an oil inlet restrictor is recommended if you have oil pressure over about 60 psig. The oil outlet should be plumbed to the oil pan above the oil level (for wet sump systems). Since the oil drain is gravity fed, it is important that the oil outlet points downward, and that the drain tube does not become horizontal or go “uphill” at any point.

Following a hot shutdown of a turbocharger, heat soak begins. This means that the heat in the head, exhaust manifold, and turbine housing finds it way to the turbo’s center housing, raising its temperature. These extreme temperatures in the center housing can result in oil coking.

To minimize the effects of heat soak-back, water-cooled center housings were introduced. These use coolant from the engine to act as a heat sink after engine shutdown, preventing the oil from coking. The water lines utilize a thermal siphon effect to reduce the peak heat soak-back temperature after key-off. The layout of the pipes should minimize peaks and troughs with the (cool) water inlet on the low side. To help this along, it is advantageous to tilt the turbocharger about 25° about the axis of shaft rotation.

Many Garrett turbos are water-cooled for enhanced durability.

Which Turbocharger is Right for Me or more affectionately known as My Turbo & Me

Selecting the proper turbocharger for your specific application requires many inputs. With decades of collective turbocharging experience, the Garrett Performance Distributors can assist in selecting the right turbocharger for your application.

The primary input in determining which turbocharger is appropriate is to have a target horsepower in mind. This should be as realistic as possible for the application. Remember that engine power is generally proportional to air and fuel flow. Thus, once you have a target power level identified, you begin to hone in on the turbocharger size, which is highly dependent on airflow requirements.

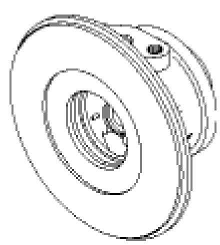
Other important factors include the type of application. An autocross car, for example, requires rapid boost response. A smaller turbocharger or smaller turbine housing would be most suitable for this application. While this will trade off ultimate power due to increased exhaust backpressure at higher engine speeds, boost response of the small turbo will be excellent.

Alternatively, on a car dedicated to track days, peak horsepower is a higher priority than low-end torque. Plus, engine speeds tend to be consistently higher. Here, a larger turbocharger or turbine housing will provide reduced backpressure but less-immediate low-end response. This is a welcome tradeoff given the intended operating conditions.

Selecting the turbocharger for your application goes beyond “how much boost” you want to run. Defining your target power level and the primary use for the application are the first steps in enabling your Garrett Performance Distributor to select the right turbocharger for you.

Bearing Housing

A grey cast iron bearing housing provides locations for a fully floating bearing system for the shaft, turbine and compressor which can rotate at speeds up to 170,000 rev/min. Shell molding is used to provide positional accuracy of critical features of the housing such as the shaft bearing and seal locations. CNC machinery mills, turns, drills and taps housing faces and connections. The bore is finish honed to meet stringent roundness, straightness and surface finish specifications.

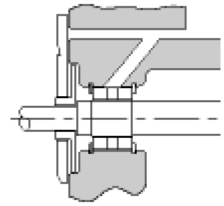


Bearing Systems

The bearing system has to withstand high temperatures, hot shut down, soot loading in the oil, contaminants, oil additives, dry starts.
Journal bearings are manufactured from specially developed bronze or brass bearing alloys. The manufacturing process is designed to create geometric tolerances and surface finishes to suit very high speed operation.

Hardened steel thrust collars and oil slingers are manufactured to strict tolerances using lapping. End thrust is absorbed in a bronze hydrodynamic thrust bearing located at the compressor end of

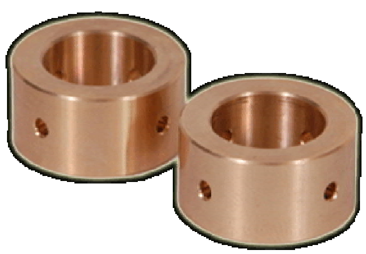
the shaft assembly. Careful sizing provides adequate load bearing capacity without excessive losses.



Journal Bearings vs. Ball Bearings

The journal bearing has long been the brawn of the turbocharger; however a ball-bearing cartridge is now an affordable technology advancement that provides significant performance improvements to the turbocharger.

Ball bearing innovation began as a result of work with the Garrett Motorsports group for several racing series where it received the term the 'cartridge ball bearing'. The cartridge is a single sleeve system that contains a set of angular contact ball bearings on either end, whereas the traditional bearing system contains a set of journal bearings and a thrust bearing



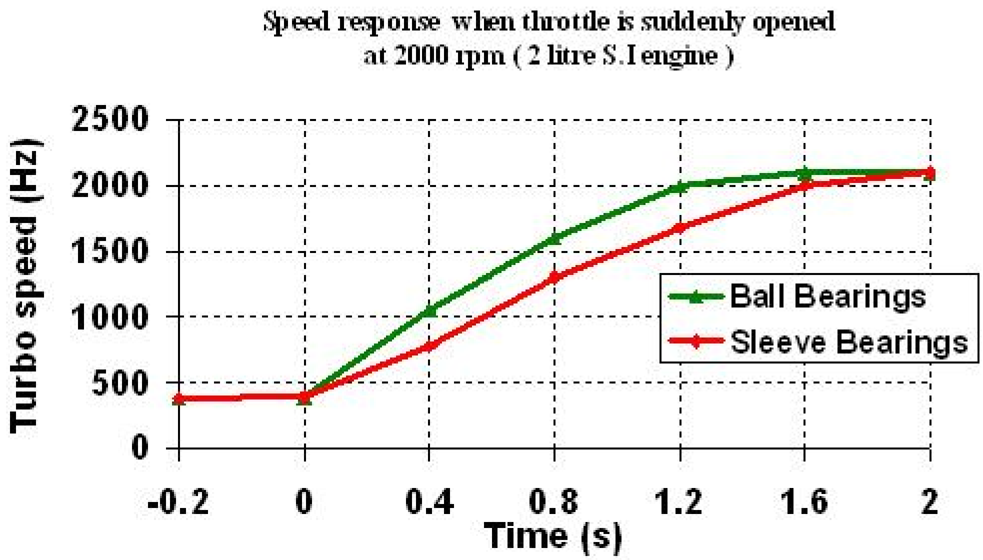
Journal Bearings

Ball Bearings

Turbo Response – When driving a vehicle with the cartridge ball bearing turbocharger, you will find exceptionally crisp and strong throttle response. Garrett Ball Bearing turbochargers spool up 15% faster than traditional journal bearings. This produces an improved response that can be converted to quicker 0-60 mph speed. In fact, some professional drivers of Garrett ball-bearing turbocharged engines report that they feel like they are driving a big, normally aspirated engine.

Tests run on CART turbos have shown that ball-bearings have up to half of the power consumption of traditional bearings. The result is faster time to boost which translates into better drivability and acceleration.

On-engine performance is also better in the steady-state for the Garrett Cartridge Ball Bearing



Reduced Oil Flow – The ball bearing design reduces the required amount of oil required to provide adequate lubrication. This lower oil volume reduces the chance for seal leakage. Also, the ball bearing is more tolerant of marginal lube conditions, and diminishes the possibility of turbocharger failure on engine shut down.

Improved Rotordynamics and Durability – The ball bearing cartridge gives better damping and control over shaft motion, allowing enhanced reliability for both everyday and extreme driving conditions. In addition, the opposed angular contact bearing cartridge eliminates the need for the thrust bearing commonly a weak link in the turbo bearing system.

Competitor Ball Bearing Options – Another option one will find is a hybrid ball bearing. This consists of replacing only the compressor side journal bearing with a single angular contact ball bearing. Since the single bearing can only take thrust in one direction, a thrust bearing is still necessary and drag in the turbine side journal bearing is unchanged. With the Garrett ball bearing cartridge the rotor-group is entirely supported by the ball bearings, maximizing efficiency, performance, and durability.

Ball Bearings in Original Equipment – Pumping up the MAZDASPEED Protégé’s heart rate is a Garrett T25 turbocharger system. With Garrett technology on board, the vehicle gains increased acceleration without sacrificing overall efficiency and it has received many rave reviews from the world’s top automotive press for it’s unprecedented performance.

Intercooler

An intercooler (also known as a **charge cooler** or an **aftercooler**) is an option to install with a turbocharger. An intercooler cools the air entering the combustion chamber, thus allowing a higher boost pressure to be run while maintaining the same intake charge temperature. When installed after the turbo it can permit substantial power gains.

Intercoolers can be either air-cooled or water-cooled, with water cooling having the greatest cooling effect, but air cooling being far cheaper and more practical. The primary purpose of intercooling is to lessen the chances of detonation due to temperature. As boost pressures rise, so do the intake charge temperatures, and thus the risk of detonation.

A secondary benefit can be had if higher boost pressures are not used in conjunction with an intercooler. By lowering the temperature of the air, the air becomes more dense and therefore contains more oxygen per unit volume. This allows for more fuel and oxygen to react in the combustion chamber, allowing for more combustion pressure and thus power to be produced. However, the small gains in cooled air can be far exceeded if the boost pressure is raised to take advantage of the cooler charge.

INTERCOOLER THEORY

An intercooler is a heat exchanger. That means there are two or more liquids or gases that don't physically touch each other but a transfer heat or energy takes place between them.

At wide open throttle and full boost the hot compressed air coming from a turbocharger is probably between 250 and 350 deg F depending on the particular turbo, boost pressure, outside air temperature, etc.. We want to cool it down, which reduces its volume so we can pack more air molecules into the cylinders and reduce the engine's likelihood of detonation.

How does an intercooler work? Hot air from the turbo flows through tubes inside the intercooler. The turbo air transfers heat to the tubes, warming the tubes and cooling the turbo air. Outside air (or water) passes over the tubes and between fins that are attached to the tubes. Heat is transferred from the hot tubes and fins to the cool outside air. This heats the outside air while cooling the tubes. This is how the turbo air is cooled down. Heat goes from the turbo air to the tubes to the outside air.

There are some useful equations which will help us understand the factors involved in transferring heat. These equations are good for any heat transfer problem, such as radiators and a/c condensers, not just intercoolers. After we look at these equations and see what's important and what's not, we can talk about what all this means.

Equation 1

The first equation describes the overall heat transfer that occurs.
Q = U x A x DTlm

Q is the amount of energy that is transferred.
U is called the heat transfer coefficient. It is a measure of how well the exchanger transfers heat. The bigger the number, the better the transfer.
A is the heat transfer area, or the surface area of the intercooler tubes and fins that is

exposed to the outside air.
DTlm is called the log mean temperature difference. It is an indication of the "driving force", or the overall average difference in temperature between the hot and cold fluids. The equation for this is:

$$DTlm = \frac{(DT1-DT2) * F}{\ln(DT1/DT2)}$$

Where **DT1** = turbo air temperature in - outside air temperature out
DT2 = turbo air temperature out - outside air temperature in
F = a correction factor, see below

Note:

The outside air that passes through the fins on the passenger side of the intercooler comes out hotter than the air passing through the fins on the drivers side of the intercooler. If you captured the air passing through all the fins and mixed it up, the temperature of this mix is the "outside air temperature out".

F is a correction factor that accounts for the fact that the cooling air coming out of the back of the intercooler is cooler on one side than the other.

To calculate this correction factor, calculate "P" and "R":

P = $\frac{\text{turbo air temp out - turbo air temp in}}{\text{outside air temp in - turbo air temp in}}$

R = $\frac{\text{outside air temp in - outside air temp out}}{\text{turbo air temp out - turbo air temp in}}$

This overall heat transfer equation shows us how to get better intercooler performance. To get colder air out of the intercooler we need to transfer more heat, or make Q bigger in other words. To make Q bigger we have to make U, A, or DTlm bigger, so that when you multiply them all together you get a bigger number. More on that later.

Caveat

These equations are all for steady state heat transfer, which we probably don't really see too much under the conditions that we are most interested in - drag race! Cruising on the highway you would definitely see steady state. Perhaps at the big end of the track you may see it too, I don't know. The material of the intercooler itself will rise in temperature when you hit full throttle, absorbing more heat than what these equations would lead you to believe. For example, at steady state idle the intercooler body may be

at 100 deg F. At steady state full throttle it may be 175 deg F. The energy it takes to heat it up to that temperature comes from the turbo outlet air, and so the cooling of that air is what is removed by both the flowing outside air and the absorption of the intercooler body. How long does it take to get to the new steady state? Beats me, but the graphs I've seen of intercooler outlet temperatures over the course of a quarter mile run lead me to believe that it is approached before you get to the end of the quarter mile, since the intercooler outlet temperatures reached a steady level.

So, now that we've got these equations, what do they really tell us?

The difference between the intercooler outlet temperature and the outside air temperature is called the approach. If it is 100 degrees outside and your intercooler cools the air going into the intake manifold down to 140 degrees, then you have an approach of 40 degrees (140 - 100 = 40). To get a better (smaller) approach you have to have more area or a better U, but there is a problem with diminishing returns. Lets rearrange the first equation to Q/DTlm = U x A. Every time DTlm goes down (get a better temperature approach) then Q goes up (transfer more heat, get a colder outlet temperature), and dividing Q by DTlm gets bigger a lot faster than U x A does. The upshot of that is we have a situation of diminishing returns; for every degree of a better approach you need more and more U x A to get there. Start with a 30 deg approach and go to 20 and you have to improve U x A by some amount, to go from 20 to 10 you need to increase U x A by an even bigger amount.

I would consider an approach of 20 degrees to be pretty good. In industrial heat exchangers it starts to get uneconomical to do better somewhere around there, the exchanger starts to get too big to justify the added expense. The only practical way of making the DTlm bigger on an existing intercooler is to only drive on cold days; if you buy a better intercooler you naturally get a better DTlm.

You can transfer more heat (and have cooler outlet temps) with more heat transfer area. That means buying a new intercooler with more tubes, more fins, longer tubes, or all three. This is what most aftermarket intercoolers strive for. Big front mounts, intercooler and a half, etc... are all increasing the area.

A practical consideration is the fin count. The area of the fins is included in the heat transfer area; more fins mean more area. If you try to pack too many fins into the intercooler the heat transfer area does go up, which is good, but the cooling air flow over the fins goes down, which is bad. Looking at the 2nd equation, Q=m*Cp*DT, when the fin count is too high then the airflow ("m") drops. For a given Q that you are trying to reach then you have to have a bigger DT, which means you have to heat up the air more. Then that affects the DTlm in the first equation, making it smaller, and lowering the overall heat transfer. So there is an optimum to be found. Starting off with bare tubes you add fins and the heat transfer goes up because you're increasing the area, and keep adding fins until it starts to choke off the cooling air flow and heat transfer starts going back down. At that point you have to add more tubes or make them longer to get more heat transfer out of the increased area.

Make U go up. You can increase the U by adding or improving "tabulators" inside the tubes. These are the fins inside the tubes which cause the air to swirl inside the tube and make it transfer its heat to the tube more efficiently. One of the best ways to increase the U is to clean the tubes out. Oil film inside the tubes acts as an insulator or thermal barrier. It keeps heat from moving from the air to the tube wall. This is expressed in our equation as a lower U. Lower U means Lower Q's which mean hotter turbo air temperatures coming out of the intercooler.

Air-to-water. If we use water as the cooling medium instead of the outside air, we can see a big improvement for several reasons: Water can absorb more energy with a lower temperature rise. This improves our DTlm, makes it bigger, which makes Q go up and outlet temps go down. A well designed water cooler exchanger also has a much bigger U, which also helps Q go up. And since both DTlm and U went up, you can make the area A smaller which makes it easier to fit the intercooler in the engine compartment. Of course there are some practical drawbacks. The need for a water circulation system is one. a big one is cooling the water down after it is heated, which means another radiator. This leads to another problem; You heat the water and cool it down with the outside air. You can't get it as cool as the outside air, but maybe you can get it within 20 degrees of it. Now you are cooling the turbo air with water that is 20 degrees hotter than the outside air, and you can only get within 15 degrees of that temperature so coming out of the intercooler you have turbo air that is 35 degrees hotter than the outside. You could have easily done that with an air to air intercooler. But if you put ice water in your holding tank and circulate that, then maybe the air temp coming out of the intercooler is 15 degrees above that or 45 to50 degrees. But after the water warms up you're back to the hot air again. Great for racing but not as good for the street.

Lower the inlet temperature. The less the turbo has to work to compress the air the lower the temperature the air coming out of the turbo is. This actually hurts DTlm, but the cooler going in the cooler coming out. You can work the turbo less by running lower boost, by improving the pressure drop between the air filter and the turbo, or by having a more efficient compressor wheel. You can also reduce the pressure drop in the intercooler, which allows you to run the same boost in the intake manifold while while having a lower turbo discharge pressure. If you can drop the turbo outlet pressure by 2 psi, or raise the turbo inlet pressure by 1 psi, that will drop the turbo discharge temperature by about 16 degrees. If the turbo air is going into the intercooler 16 degrees colder then it may come out only 10 degrees colder than before, but that is still better than it was.

What about my Intercooler?

Wondering if your intercooler is up to snuff? The big test: measure your intercooler outlet temperature! When I did this I got a K type thermocouple, the thin wire kind, slid it under the throttle body/up pipe hose and down into the center of the up pipe, and went for a drive. On an 80 to 85 deg day I got a WOT temperature of 140 deg, for a 55 to 60 deg approach. That tells me that I need more intercooler. If I can get the temperature down to 100 deg, the air density in the intake manifold goes up by 7%, so I

should flow 7% more air and presumably make 7% more hp. On a 350 hp engine that is 25 hp increase. On a 450 hp engine that's a 30 hp increase. Damn, where's my check book…

Another check is pressure drop. Best way to check it is to find a pressure differential gauge, which has 2 lines instead of the single line a normal pressure gauge has. It checks the difference between the 2 spots it is hooked up to, as opposed to checking the difference in pressure between the spot it is hooked up to and atmospheric pressure, which is how a normal pressure gauge works.

Hook one line of the gauge to the turbo outlet and one to (preferably) the intercooler outlet. The turbo outlet/intercooler inlet pressure is easy, just tee into the wastegate supply line off the compressor housing. It would be nice to get the intercooler outlet pressure directly, but there's no convenient spot to hook up to. Hooking into the intake manifold (such as via the line to the boost gauge) is quite convenient, but gives the total pressure drop: intercooler + up pipe + throttle body. That'll give you a pretty good idea though.

Instead of the differential pressure gauge you could use 2 boost gauges, one in each spot, but then you have to worry about whether both gauges are calibrated the same, try to read both at the same time while driving fast, etc AND you may spring (i.e., ruin) the gauge on the turbo outlet since when you close the throttle you get a big pressure spike that your normal boost gauge never sees.

If you find more than 4 or 5 psi difference between the intercooler inlet and intake manifold (and I'm just giving an educated guess here, you'd probably want to refer to one of the intercooler manufacturers for a better number) then I would suspect that a larger, lower pressure drop intercooler would offer you some gains.

TURBO TYPES

The turbocharger turbine, which consists of a turbine wheel and a turbine housing, converts the engine exhaust gas into mechanical energy to drive the compressor. The gas, which is restricted by the turbine's flow cross-sectional area, results in a pressure and temperature drop between the inlet and outlet. This pressure drop is converted by the turbine into kinetic energy to drive the turbine wheel.

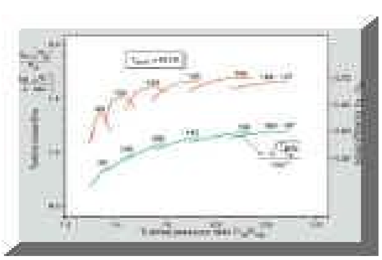
There are two main turbine types: axial and radial flow. In the axial-flow type, flow through the wheel is only in the axial direction. In radial-flow turbines, gas inflow is centripetal, i.e. in a radial direction from the outside in, and gas outflow in an axial direction.

Up to a wheel diameter of about 160 mm, only radial-flow turbines are used. This corresponds to an engine power of approximately 1000 kW per turbocharger. From 300 mm onwards, only axial-flow turbines are used. Between these two values, both variants are possible.

As the radial-flow turbine is the most popular type for automotive applications, the following description is limited to the design and function of this turbine type. In the volute of such radial or centripetal turbines, exhaust gas pressure is converted into kinetic energy and the exhaust gas at the wheel circumference is directed at constant velocity to the turbine wheel. Energy transfer from kinetic energy into shaft power takes place in the turbine wheel, which is designed so that nearly all the kinetic energy is converted by the time the gas reaches the wheel outlet.

Operating characteristics

The turbine performance increases as the pressure drop between the inlet and outlet increases, i.e. when more exhaust gas is dammed upstream of the turbine as a result of a higher engine speed, or in the case of an exhaust gas temperature rise due to higher exhaust gas energy.



Turbocharger turbine map

The turbine's characteristic behaviors is determined by the specific flow cross-section, the throat cross-section, in the transition area of the inlet channel to the volute. By reducing this throat cross-section, more exhaust gas is dammed upstream of the turbine and the turbine performance increases as a result of the higher pressure ratio. A smaller flow cross-section therefore results in higher boost pressures.
The turbine's flow cross-sectional area can be easily varied by changing the turbine housing.

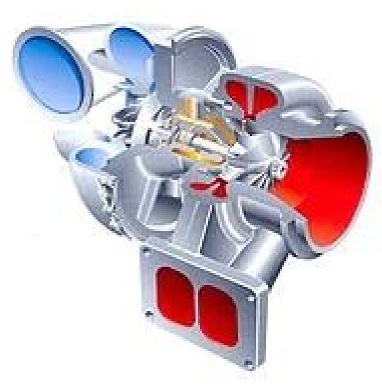
Besides the turbine housing flow cross-sectional area, the exit area at the wheel inlet also influences the turbine's mass flow capacity. The machining of a turbine wheel cast contour allows the cross-sectional area and, therefore, the boost pressure, to be adjusted. A contour enlargement results in a larger flow cross-sectional area of the turbine.

Turbines with variable turbine geometry change the flow cross-section between volute channel and wheel inlet. The exit area to the turbine wheel is changed by variable guide vanes or a variable sliding ring covering a part of the cross-section.

In practice, the operating characteristics of exhaust gas turbocharger turbines are described by maps showing the flow parameters plotted against the turbine pressure ratio. The turbine map shows the mass flow curves and the turbine efficiency for various speeds. To simplify the map, the mass flow curves, as well as the efficiency, can be shown by a mean curve

For a high overall turbocharger efficiency, the co-ordination of compressor and turbine wheel diameters is of vital importance. The position of the operating point on the compressor map determines the turbocharger speed. The turbine wheel diameter has to be such that the turbine efficiency is maximized in this operating range.

Twin-entry turbines

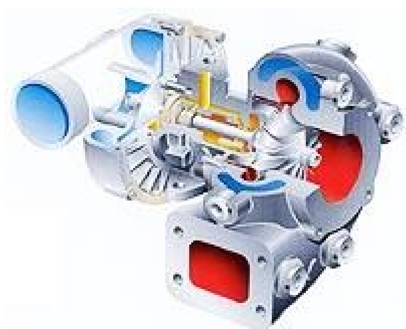


Turbocharger with twin-entry turbine

The turbine is rarely subjected to constant exhaust pressure. In pulse turbocharged commercial diesel engines, twin-entry turbines allow exhaust gas pulsations to be optimized, because a higher turbine pressure ratio is reached in a shorter time. Thus, through the increasing pressure ratio, the efficiency rises, improving the all-important time interval when a high, more efficient mass flow is passing through the turbine. As a result of this improved exhaust gas energy utilization, the engine's boost pressure characteristics and, hence, torque behavior is improved, particularly at low engine speeds.

To prevent the various cylinders from interfering with each other during the charge exchange cycles, three cylinders are connected into one exhaust gas manifold. Twin-entry turbines then allow the exhaust gas flow to be fed separately through the turbine.

Water-cooled turbine housings



Turbocharger with water-cooled turbine housing for marine applications

Safety aspects also have to be taken into account in turbocharger design. In ship engine rooms, for instance, hot surfaces have to be avoided because of fire risks. Therefore, water-cooled turbocharger turbine housings or housings coated with insulating material are used for marine applications.

Please thoroughly review and have a good understanding of Turbo Systems - Basic prior to reading this section. The following areas will be covered in the Turbo System - Advanced section:

1. Wheel trim topic coverage

2. Understanding turbine housing A/R and housing sizing

3. Different types of manifolds (advantages/disadvantages log style vs. equal length)

4. Compression ratio with boost

5. Air/Fuel Ratio tuning: Rich v. Lean, why lean makes more power but is more dangerous

1. Wheel trim topic coverage

Trim is a common term used when talking about or describing turbochargers. For example, you may hear someone say "I have a GT2871R ' **56 Trim** ' turbocharger. What is 'Trim?' Trim is a term to express the relationship between the inducer* and exducer* of both turbine and compressor wheels. More accurately, it is an area ratio.

* The inducer diameter is defined as the diameter where the air enters the wheel, whereas the exducer diameter is defined as the diameter where the air exits the wheel.

Based on aerodynamics and air entry paths, the inducer for a compressor wheel is the smaller diameter. For turbine wheels, the inducer it is the larger diameter (see Figure 1.)

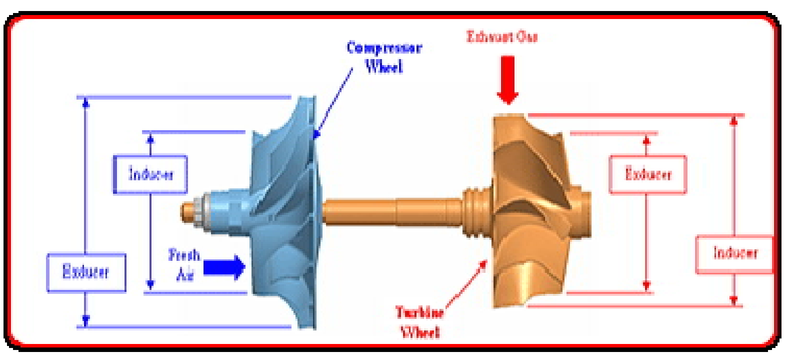


Figure 1. Illustration of the inducer and exducer diameter of compressor and turbine wheels

Example #1: GT2871R turbocharger (Garrett part number 743347-2) has a compressor wheel with the below dimensions. What is the trim of the compressor wheel?

Inducer diameter = 53.1mm
Exducer diameter = 71.0mm

$$Trim = \left(\frac{Inducer^2}{Exducer^2}\right)*100$$

$$Trim = \left(\frac{53.1^2}{71.0^2}\right)*100$$

$$Trim = 56$$

Example #2: GT2871R turbocharger (part # 743347-1) has a compressor wheel with an exducer diameter of 71.0mm and a trim of 48. What is the inducer diameter of the compressor wheel?

Exducer diameter = 71.0mm
Trim = 48

$$Trim = \left(\frac{Inducer^2}{Exducer^2}\right)*100$$

$$Inducer^2 = \left(\frac{Trim}{100}\right)*Exducer^2$$

$$Inducer = \sqrt{Trim/100}*Exducer$$

$$Inducer = \sqrt{48/100}*71.0$$

$$Inducer = 49.2mm$$

The trim of a wheel, whether compressor or turbine, affects performance by shifting the airflow capacity. All other factors held constant, a higher trim wheel will flow more than a smaller trim wheel.

However, it is important to note that very often all other factors are not held constant. So just because a wheel is a larger trim does not necessarily mean that it will flow more.

2. Understanding housing sizing: A/R

A/R (Area/Radius) describes a geometric characteristic of all compressor and turbine housings. Technically, it is defined as:

the inlet (or, for compressor housings, the discharge) cross-sectional area divided by the radius from the turbo centerline to the centroid of that area (see Figure 2.).

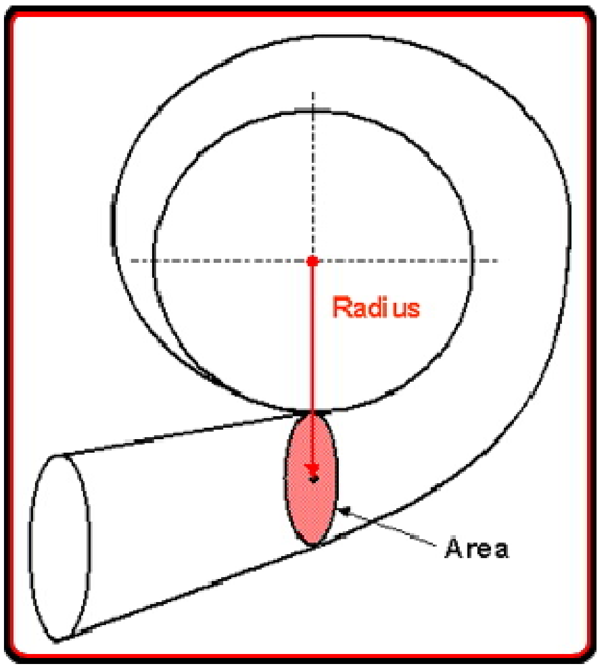


Figure 2. Illustration of compressor housing showing A/R characteristic

The A/R parameter has different effects on the compressor and turbine performance, as outlined below.

Compressor A/R - Compressor performance is comparatively insensitive to changes in A/R. Larger A/R housings are sometimes used to optimize performance of low boost applications, and smaller A/R are used for high boost applications. However, as this influence of A/R on compressor performance is minor, there are not A/R options available for compressor housings.

Turbine A/R - Turbine performance is greatly affected by changing the A/R of the housing, as it is used to adjust the flow capacity of the turbine. Using a smaller A/R will increase the exhaust gas velocity into the turbine wheel. This provides increased turbine power at lower engine speeds, resulting in a quicker boost rise. However, a small A/R also causes the flow to enter the wheel more tangentially, which reduces the ultimate flow capacity of the turbine wheel. This will tend to increase exhaust backpressure and hence reduce the engine's ability to "breathe" effectively at high RPM, adversely affecting peak engine power.

Conversely, using a larger A/R will lower exhaust gas velocity, and delay boost rise. The flow in a larger A/R housing enters the wheel in a more radial fashion, increasing the wheel's effective flow capacity, resulting in lower backpressure and better power at higher engine speeds.

When deciding between A/R options, be realistic with the intended vehicle use and use the A/R to bias the performance toward the desired powerband characteristic.

Here's a simplistic look at comparing turbine housing geometry with different applications. By comparing different turbine housing A/R, it is often possible to determine the intended use of the system.

Imagine two 3.5L engines both using GT30R turbochargers. The only difference between the two engines is a different turbine housing A/R; otherwise the two engines are identical:

1. Engine #1 has turbine housing with an A/R of 0.63
2. Engine #2 has a turbine housing with an A/R of 1.06.

What can we infer about the intended use and the turbocharger matching for each engine?

Engine#1: This engine is using a smaller A/R turbine housing (0.63) thus biased more towards low-end torque and optimal boost response. Many would describe this as being more "fun" to drive on the street, as normal daily driving habits tend to favor transient response. However, at higher engine speeds, this smaller A/R housing will result in high backpressure, which can result in a loss of top end power. This type of engine performance is desirable for street applications where the low speed boost response and transient conditions are more important than top end power.

Engine #2: This engine is using a larger A/R turbine housing (1.06) and is biased towards peak horsepower, while sacrificing transient response and torque at very low engine speeds. The larger A/R turbine housing will continue to minimize backpressure at high rpm, to the benefit of engine peak power. On the other hand, this will also raise the engine speed at which the turbo can provide boost, increasing time to boost. The performance of Engine #2 is more desirable for racing applications than Engine #1 where the engine will be operating at high engine speeds most of the time.

3. Different types of manifolds (advantages/disadvantages log style vs. equal length)

There are two different types of turbocharger manifolds; cast log style (see Figure 3.) and welded tubular style (see Figure 4.).

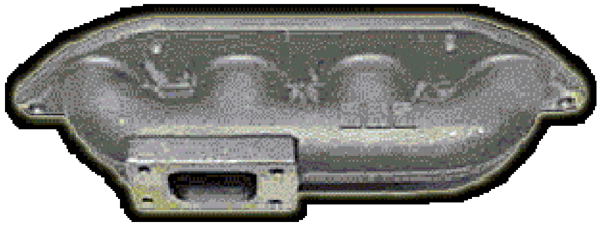


Figure 3. Cast log style turbocharger manifold

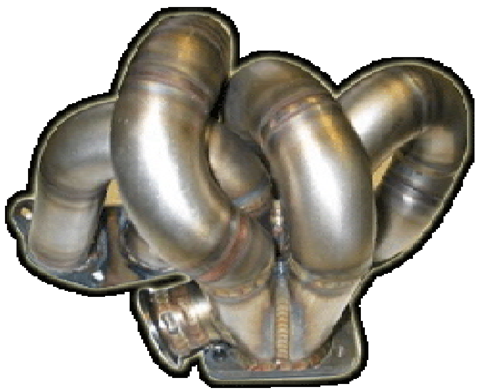


Figure 4. Welded tubular turbocharger manifold

Manifold design on turbocharged applications is deceptively complex as there many factors to take into account and trade off

General design tips for best overall performance are to:

- Maximize the radius of the bends that make up the exhaust primaries to maintain pulse energy
- Make the exhaust primaries equal length to balance exhaust reversion across all cylinders
- Avoid rapid area changes to maintain pulse energy to the turbine
- At the collector, introduce flow from all runners at a narrow angle to minimize "turning" of the flow in the collector
- For better boost response, minimize the exhaust volume between the exhaust ports and the turbine inlet
- For best power, tuned primary lengths can be used

Cast manifolds are commonly found on OEM applications, whereas welded tubular manifolds are found almost exclusively on aftermarket and race applications. Both manifold types have their advantages and disadvantages. Cast manifolds are generally very durable and are usually dedicated to one application. They require special tooling for the casting and machining of specific features on the manifold. This tooling can be expensive.

On the other hand, welded tubular manifolds can be custom-made for a specific application without special tooling requirements. The manufacturer typically cuts pre-bent steel U-bends into the desired geometry and then welds all of the components together. Welded tubular manifolds are a very effective solution. One item of note is durability of this design. Because of the welded joints, thinner wall sections, and reduced stiffness, these types of manifolds are often susceptible to cracking due to thermal expansion/contraction and vibration. Properly constructed tubular manifolds can last a long time, however. In addition, tubular manifolds can offer a substantial performance advantage over a log-type manifold.

A design feature that can be common to both manifold types is a " **DIVIDED MANIFOLD"** , typically employed with " **DIVIDED** " or "twin-scroll" turbine housings. Divided exhaust manifolds can be incorporated into either a cast or welded tubular manifolds (see Figure 5. and Figure 6.).

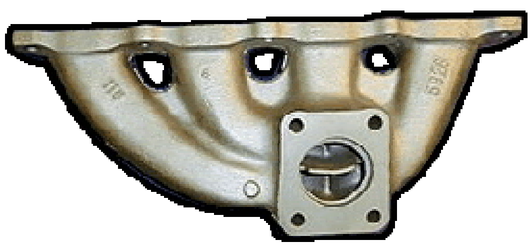


Figure 5. Cast manifold with a divided turbine inlet design feature

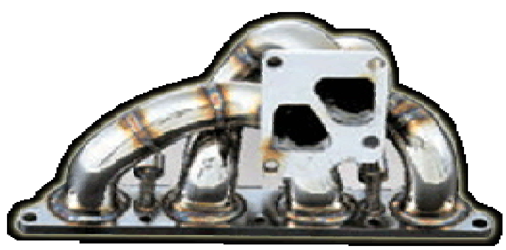


Figure 6. Welded tubular manifold with a divided turbine inlet design feature

The concept is to DIVIDE or separate the cylinders whose cycles interfere with one another to best utilize the engine's exhaust pulse energy.

For example, on a four-cylinder engine with firing order 1-3-4-2, cylinder #1 is ending its expansion stroke and opening its exhaust valve while cylinder #2 still has its exhaust valve open (cylinder #2 is in its overlap period). In an undivided exhaust manifold, this pressure pulse from cylinder #1's exhaust blowdown event is much more likely to contaminate cylinder #2 with high pressure exhaust gas. Not only does this hurt cylinder #2's ability to breathe properly, but this pulse energy would have been better utilized in the turbine.

The proper grouping for this engine is to keep complementary cylinders grouped together-- #1 and #4 are complementary; as are cylinders #2 and #3.

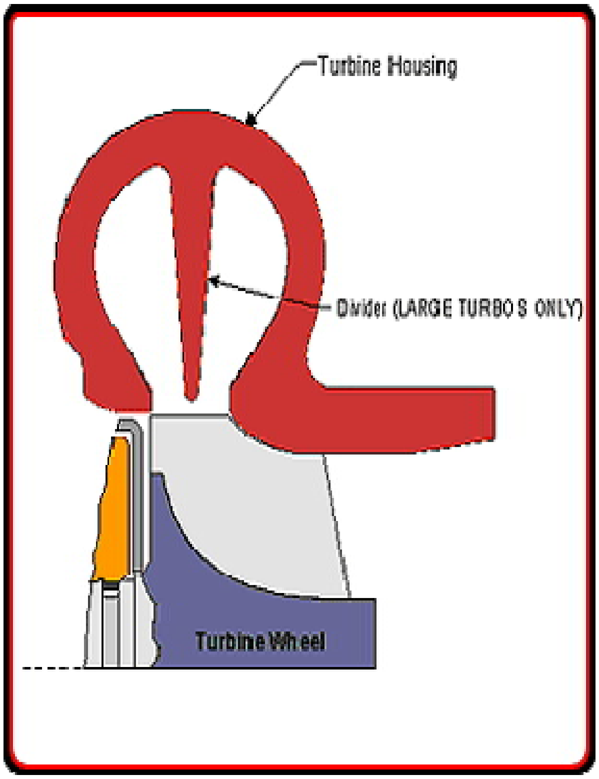


Figure 7. Illustration of divided turbine housing

Because of the better utilization of the exhaust pulse energy, the turbine's performance is improved and boost increases more quickly.

4. Compression ratio with boost

Before discussing compression ratio and boost, it is important to understand engine knock, also known as detonation. Knock is a dangerous condition caused by uncontrolled combustion of the air/fuel mixture. This abnormal combustion causes rapid spikes in cylinder pressure which can result in engine damage.

Three primary factors that influence engine knock are:

1. **Knock resistance characteristics (knock limit) of the engine:** Since every engine is vastly different when it comes to knock resistance, there is no single answer to "how much." Design features such as combustion chamber geometry, spark plug location, bore size and compression ratio all affect the knock characteristics of an engine.

2. **Ambient air conditions:** For the turbocharger application, both ambient air conditions and engine inlet conditions affect maximum boost. Hot air and high cylinder pressure increases the tendency of an engine to knock. When an engine is boosted, the intake air temperature increases, thus increasing the tendency to knock. Charge air cooling (e.g. an intercooler) addresses this concern by cooling the compressed air produced by the turbocharger
3. **Octane rating of the fuel being used:** octane is a measure of a fuel's ability to resist knock. The octane rating for pump gas ranges from 85 to 94, while racing fuel would be well above 100. The higher the octane rating of the fuel, the more resistant to knock. Since knock can be damaging to an engine, it is important to use fuel of sufficient octane for the application. Generally speaking, the more boost run, the higher the octane requirement.

This cannot be overstated: engine calibration of fuel and spark plays an enormous role in dictating knock behavior of an engine. See Section 5 below for more details.

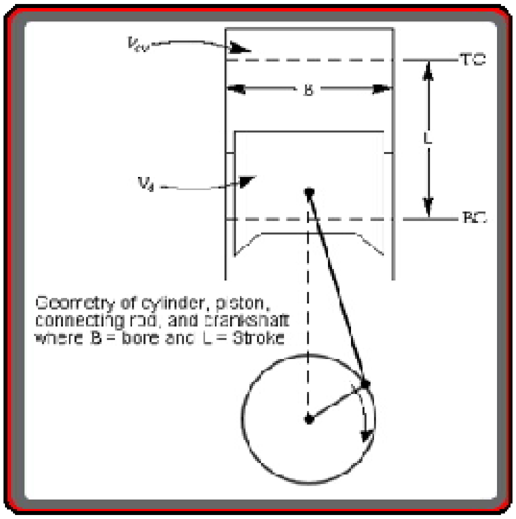
Now that we have introduced knock/detonation, contributing factors and ways to decrease the likelihood of detonation, let's talk about compression ratio. Compression ratio is defined as:

$$Compression\_Ratio = \frac{displacement\_volume + clearance\_volume}{clearance\_volume}$$

or

$$CR = \frac{V_d + V_{cv}}{V_{cv}}$$

where
CR = compression ratio
V_d = displacement volume
V_{cv} = clearance volume



The compression ratio from the factory will be different for naturally aspirated engines and boosted engines. For example, a stock Honda S2000 has a compression ratio of 11.1:1, whereas a turbocharged Subaru Impreza WRX has a compression ratio of 8.0:1.

There are numerous factors that affect the maximum allowable compression ratio. There is no single correct answer for every application. Generally, compression ratio should be set as high as feasible without encountering detonation at the maximum load condition. Compression ratio that is too low will result in an engine that is a bit sluggish in off-boost operation. However, if it is too high this can lead to serious knock-related engine problems.

Factors that influence the compression ratio include: fuel anti-knock properties (octane rating), boost pressure, intake air temperature, combustion chamber design, ignition timing, valve events, and exhaust backpressure. Many modern normally-aspirated engines have well-designed combustion chambers that, with appropriate tuning, will allow modest boost levels with no change to compression ratio. For higher power targets with more boost , compression ratio should be adjusted to compensate.

There are a handful of ways to reduce compression ratio, some better than others. Least desirable is adding a spacer between the block and the head. These spacers reduce the amount a "quench" designed into an engine's combustion chambers, and can alter cam timing as well. Spacers are, however, relatively simple and inexpensive.

A better option, if more expensive and time-consuming to install, is to use lower-compression pistons. These will have no adverse effects on cam timing or the head's ability to seal, and allow proper quench regions in the combustion chambers.

5. Air/Fuel Ratio tuning: Rich v. Lean, why lean makes more power but is more dangerous

When discussing engine tuning the **'Air/Fuel Ratio'** (AFR) is one of the main topics. Proper AFR calibration is critical to performance and durability of the engine and it's components. The AFR defines the ratio of the amount of air consumed by the engine compared to the amount of fuel.

A **'Stoichiometric'** AFR has the correct amount of air and fuel to produce a chemically complete combustion event. For gasoline engines, the stoichiometric , A/F ratio is 14.7:1, which means 14.7 parts of air to one part of fuel. The stoichiometric AFR depends on fuel type-- for alcohol it is 6.4:1 and 14.5:1 for diesel.

So what is meant by a rich or lean AFR? A lower AFR number contains less air than the 14.7:1 stoichiometric AFR, therefore it is a richer mixture. Conversely, a higher AFR number contains more air and therefore it is a leaner mixture.

For Example:
15.0:1 = Lean
14.7:1 = Stoichiometric
13.0:1 = Rich

Leaner AFR results in higher temperatures as the mixture is combusted. Generally, normally-aspirated spark-ignition (SI) gasoline engines produce maximum power just slightly rich of stoichiometric. However, in practice it is kept between 12:1 and 13:1 in order to keep exhaust gas temperatures in check and to account for variances in fuel quality. This is a realistic full-load AFR on a normally-aspirated engine but can be dangerously lean with a highly-boosted engine.

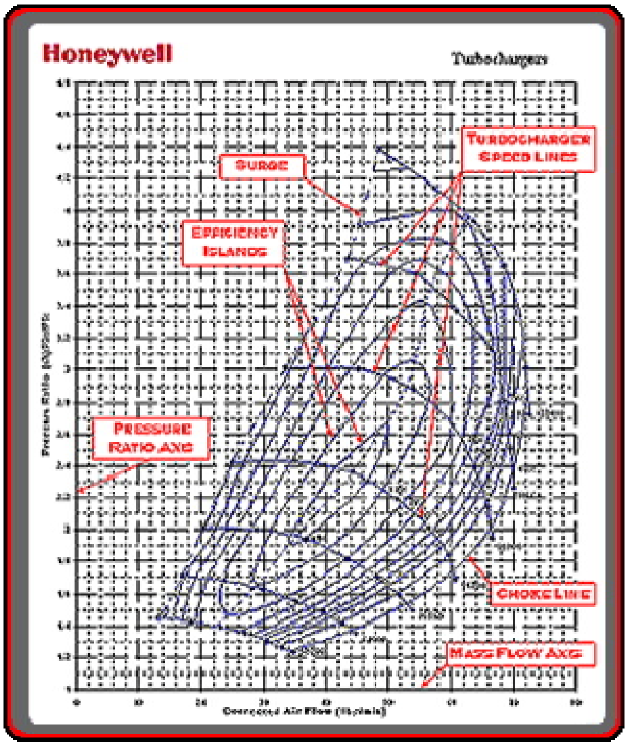
Let's take a closer look. As the air-fuel mixture is ignited by the spark plug, a flame front propagates from the spark plug. The now-burning mixture raises the cylinder pressure and temperature, peaking at some point in the combustion process.

The turbocharger increases the density of the air resulting in a denser mixture. The denser mixture raises the peak cylinder pressure, therefore increasing the probability of knock. As the AFR is leaned out, the temperature of the burning gases increases, which also increases the probability of knock. This is why it is imperative to run richer AFR on a boosted engine at full load. Doing so will reduce the likelihood of knock, and will also keep temperatures under control.

There are actually three ways to reduce the probability of knock at full load on a turbocharged engine: reduce boost, adjust the AFR to richer mixture, and retard ignition timing. These three parameters need to be optimized together to yield the highest reliable power.

Parts of the Compressor Map:

◊ The compressor map is a graph that describes a particular compressor's performance characteristics, including efficiency, mass flow range, boost pressure capability, and turbo speed. Shown below is a figure that identifies aspects of a typical compressor map:



◊ Pressure Ratio

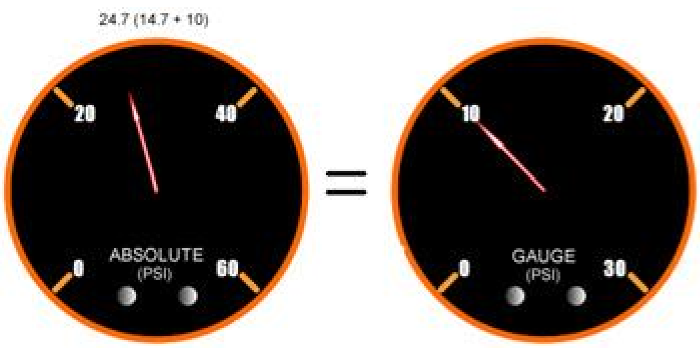
- Pressure Ratio (Πc) is defined as the Absolute outlet pressure divided by the Absolute inlet pressure.

  $$\Pi c = {P_{2c}}/{P_{1c}}$$

  Where:
  - Πc = Pressure Ratio
  - P_{2c} = Compressor
  - Discharge Pressure
  - P_{1c} = Compressor Inlet Pressure
- It is important to use units of **Absolute Pressure** for both P1c and P2c. Remember that Absolute Pressure at sea level is 14.7 psia (in units of psia, the a refers to "absolute"). This is referred to as standard atmospheric pressure at standard conditions.

- **Gauge Pressure** (in units of psig, the g refers to "gauge") measures the pressure *above* atmospheric, so a gauge pressure reading at atmospheric conditions will read zero. Boost gauges measure the manifold pressure relative to atmospheric pressure, and thus are measuring Gauge Pressure. This is important when determining P2c. For example, a reading of 12 psig on a boost gauge means that the air pressure in the manifold is 12 psi above atmospheric pressure. For a day at standard atmospheric conditions,

12 psig + 14.7 psia = 26.7 psi absolute pressure in the manifold



- The **pressure ratio** at this condition can now be calculated:

26.7 psia / 14.7 psia = **1.82**

- However, this assumes there is no adverse impact of the air filter assembly at the compressor inlet.
- In determining pressure ratio, the absolute pressure at the compressor inlet (P_{2c}) is often LESS than the ambient pressure, especially at high load. Why is this? Any restriction (caused by the air filter or restrictive ducting) will result in a "depression," or pressure loss, upstream of the compressor that needs to be accounted for when determining pressure ratio. This depression can be 1 psig or more on some intake systems. In this case P_{1c} on a standard day is:

14.7psia – 1 psig = 13.7 psia at compressor inlet

- Taking into account the 1 psig intake depression, the **pressure ratio** is now:

(12 psig + 14.7 psia) / 13.7 psia = **1.95.**

- That's great, but what if you're not at sea level? In this case, simply substitute the actual atmospheric pressure in place of the 14.7 psi in the equations above to give a more accurate calculation. At higher elevations, this can have a significant effect on pressure ratio.

  For example, at Denver's 5000 feet elevation, the atmospheric pressure is typically around 12.4 psia. In this case, the **pressure ratio** calculation, taking into account the intake depression, is:

  $$(12 \text{ psig} + 12.4 \text{ psia}) / (12.4 \text{ psia} – 1 \text{ psig}) = \mathbf{2.14}$$

  Compared to the 1.82 pressure ratio calculated originally, this is a big difference.

- As you can see in the above examples, pressure ratio depends on a lot more than just boost.

◊ Mass Flow Rate

- Mass Flow Rate is the mass of air flowing through a compressor (and engine!) over a given period of time and is commonly expressed as lb/min (pounds per minute). Mass flow can be physically measured, but in many cases it is sufficient to estimate the mass flow for choosing the proper turbo.
- Many people use Volumetric Flow Rate (expressed in cubic feet per minute, CFM or ft^3/min) instead of mass flow rate. Volumetric flow rate can be converted to mass flow by multiplying by the air density. Air density at sea level is 0.076lb/ft^3
- What is my mass flow rate? As a very general rule, turbocharged gasoline engines will generate 9.5-10.5 horsepower (as measured at the flywheel) for each lb/min of airflow. So, an engine with a target peak horsepower of 400 Hp will require 36-44 lb/min of airflow to achieve that target. This is just a rough first approximation to help narrow the turbo selection options.

◊ Surge Line

- Surge is the left hand boundary of the compressor map. Operation to the left of this line represents a region of flow instability. This region is characterized by mild flutter to wildly fluctuating boost and "barking" from the compressor. Continued operation within this region can lead to premature turbo failure due to heavy thrust loading.
- Surge is most commonly experienced when one of two situations exist. The first and most damaging is surge under load. It can be an indication that your compressor is too large. Surge is also commonly experienced when the throttle is quickly closed after boosting. This occurs because mass flow is drastically reduced as the throttle is closed, but the turbo is still spinning and generating

boost. This immediately drives the operating point to the far left of the compressor map, right into surge.

Surge will decay once the turbo speed finally slows enough to reduce the boost and move the operating point back into the stable region. This situation is commonly addressed by using a Blow-Off Valves (BOV) or bypass valve. A BOV functions to vent intake pressure to atmosphere so that the mass flow ramps down smoothly, keeping the compressor out of surge. In the case of a recirculating bypass valve, the airflow is recirculated back to the compressor inlet.

- A Ported Shroud compressor (see Fig. 2) is a feature that is incorporated into the compressor housing. It functions to move the surge line further to the left (see Fig. 3) by allowing some airflow to exit the wheel through the port to keep surge from occurring. This provides additional useable range and allows a larger compressor to be used for higher flow requirements without risking running the compressor into a dangerous surge condition. The presence of the ported shroud usually has a minor negative impact on compressor efficiency.

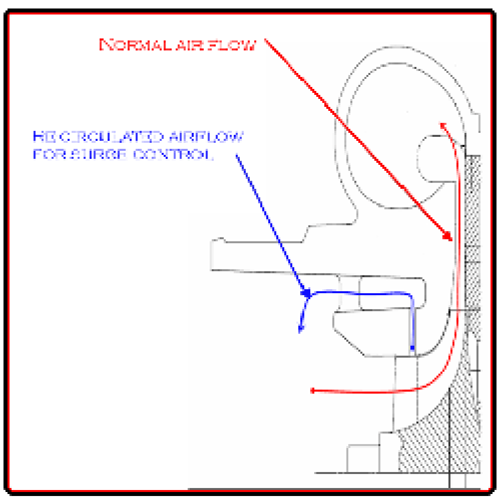


Fig. 2

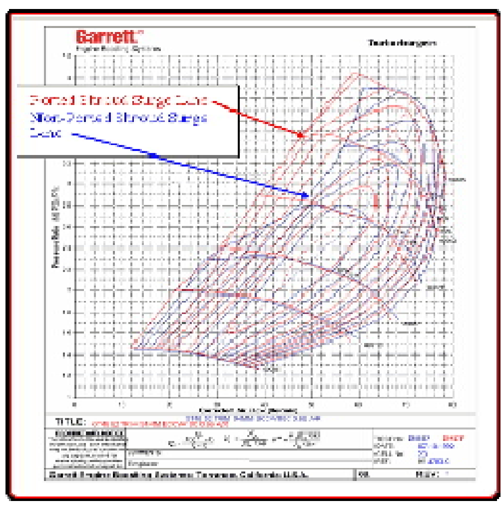


Fig. 3

◊ **The Choke Line** is the right hand boundary of the compressor map. For Garrett maps, the choke line is typically defined by the point where the efficiency drops below 58%. In addition to the rapid drop of compressor efficiency past this point, the turbo speed will also be approaching or exceeding the allowable limit. If your actual or predicted operation is beyond this limit, a larger compressor is necessary.

◊ **Turbo Speed Lines** are lines of constant turbo speed. Turbo speed for points between these lines can be estimated by interpolation. As turbo speed increases, the pressure ratio increases and/or mass flow increases. As indicated above in the choke line description, the turbo speed lines are very close together at the far right edge of the map. Once a compressor is operating past the choke limit, turbo speed increases very quickly and a turbo over-speed condition is very likely.

◊ **Efficiency Islands** are concentric regions on the maps that represent the compressor efficiency at any point on the map. The smallest island near the center of the map is the highest or peak efficiency island. As the rings move out from there, the efficiency drops by the indicated amount until the surge and choke limits are reached.

Plotting Your Data on the Compressor Map

In this section, methods to calculate mass flow rate and boost pressure required to meet a horsepower target are presented. This data will then be used to choose the appropriate compressor and turbocharger. Having a horsepower target in mind is a vital part of the process. In addition to being necessary for calculating mass flow and boost pressure, a horsepower target is required for choosing the right fuel injectors, fuel pump and regulator, and other engine components.

◊ Estimating Required Air Mass Flow and Boost Pressures to reach a Horsepower target.

· Things you need to know:
- **Horsepower Target**
- **Engine displacement**
- **Maximum RPM**
- **Ambient conditions** (temperature and barometric pressure. Barometric pressure is usually given as inches of mercury and can be converted to psi by dividing by 2)

· Things you need to estimate:

· Engine Volumetric Efficiency. Typical numbers for peak Volumetric Efficiency (VE) range in the 95%-99% for modern 4-valve heads, to 88% -95% for 2-valve designs. If you have a torque curve for your engine, you can use this to estimate VE at various engine speeds. On a well-tuned engine, the VE will peak at the torque peak, and this number can be used to scale the VE at other engine speeds. A 4-valve engine will typically have higher VE over more of its rev range than a two-valve engine.

· Intake Manifold Temperature. Compressors with higher efficiency give lower manifold temperatures. Manifold temperatures of intercooled setups are typically 100 - 130 degrees F, while non-intercooled values can reach from 175-300 degrees F.

· Brake Specific Fuel Consumption (BSFC). BSFC describes the fuel flow rate

required to generate each horsepower. General values of BSFC for turbocharged gasoline engines range from 0.50 to 0.60 and higher. The units of BSFC are $\frac{lb}{Hp*hr}$ Lower BSFC means that the engine requires less fuel to generate a given horsepower. Race fuels and aggressive tuning are required to reach the low end of the BSFC range described above.

For the equations below, we will divide BSFC by 60 to convert from hours to minutes.

To plot the compressor operating point, first calculate airflow:

$$Wa = HP * A/F * BSFC/60$$

Where:
· Wa = $Airflow_{actual}$ (lb/min)
· HP = Horsepower Target (flywheel)
· A/F = Air/Fuel Ratio
· $BSFC/60$ = Brake Specific Fuel Consumption ($\frac{lb}{Hp*hr}$) ÷ 60 (to convert from hours to minutes)

EXAMPLE:

I have an engine that I would like to use to make 400Hp, I want to choose an air/fuel ratio of 12 and use a BSFC of 0.55. Plugging these numbers into the formula from above:

$$Wa = 400 * 12 * 0.55/60 = 44.0\ lb/\min$$ of air.

Thus, a compressor map that has the capability of at least 44 pounds per minute of airflow capacity is a good starting point.

Note that nowhere in this calculation did we enter any engine displacement or RPM numbers. This means that for any engine, in order to make 400 Hp, it needs to flow about 44 lb/min (this assumes that BSFC remains constant across all engine types).

Naturally, a smaller displacement engine will require more boost or higher engine speed to meet this target than a larger engine will. So how much boost pressure would be required?

◊ Calculate required manifold pressure required to meet the horsepower, or flow target:

$$MAP_{req} = \frac{Wa * R * (460 + T_m)}{VE * N/2 * Vd}$$

Where:

- MAP_{req} = Manifold Absolute Pressure (psia) required to meet the horsepower target
- Wa = $Airflow_{actual}$(lb/min)
- R = Gas Constant = 639.6
- Tm = Intake Manifold Temperature (degrees F)
- VE = Volumetric Efficiency
- N = Engine speed (RPM)
- Vd = engine displacement (Cubic Inches, convert from liters to CI by multiplying by 61.02, ex. 2.0 liters * 61.02 = 122 CI)

EXAMPLE:

To continue the example above, let's consider a 2.0 liter engine with the following description:

- Wa = 44 lb/min as previously calculated
- Tm = 130 degrees F
- VE = 92% at peak power
- N = 7200 RPM
- Vd = 2.0 liters * 61.02 = 122 CI

$$MAP_{req} = \frac{44 * 639.6 * (460 + 130)}{.92 * 7200/2 * 122}$$

= **41.1 psia** (remember, this is absolute pressure. Subtract atmospheric pressure to get gauge pressure (aka boost):

41.1 psia – 14.7 psia (at sea level) = 26.4 psig boost

As a comparison let's repeat the calculation for a larger displacement 5.0L (4942 cc/302 CI) engine.

Where:

- Wa = 44 lb/min as previously calculated
- Tm = 130 degrees F
- VE = 85% at peak power (it is a pushrod V-8)

· N = 6000 RPM
· Vd = 4.942*61.02= 302 CI

$$MAP_{req} = \frac{44 * 639.6 * (460 + 130)}{.85 * 6000/2 * 302}$$

= **21.6 psia** (or 6.9 psig boost)

This example illustrates in order to reach the horsepower target of 400 hp, a larger engine requires lower manifold pressure *but still needs 44lb/min of airflow*. This can have a very significant effect on choosing the correct compressor.

With Mass Flow and Manifold Pressure, we are nearly ready to plot the data on the compressor map. The next step is to determine how much pressure loss exists between the compressor and the manifold. The best way to do this is to measure the pressure drop with a data acquisition system, but many times that is not practical.

Depending upon flow rate, charge air cooler characteristics, piping size, number/quality of the bends, throttle body restriction, etc., the plumbing pressure drop can be estimated. This can be 1 psi or less for a very well designed system. On certain restrictive OEM setups, especially those that have now higher-than-stock airflow levels, the pressure drop can be 4 psi or greater.

For our examples we will assume that there is a 2 psi loss. So to determine the Compressor Discharge Pressure (P_{2c}), 2 psi will be added to the manifold pressure calculated above.

$$P_{2c} = MAP + \Delta P_{loss}$$

Where:

· P_{2c} = Compressor Discharge Pressure (psia)
· MAP = Manifold Absolute Pressure (psia)
· ΔP_{loss} = Pressure Loss Between the Compressor and the Manifold (psi)

For the 2.0 L engine:

$$P_{2c} = 41.1 + 2$$

= 43.1 psia

For the 5.0 L engine:

$P_{2c} = 21.6 + 2$

= 23.6 psia

Remember our discussion on inlet depression in the Pressure Ratio discussion earlier, we said that a typical value might be 1 psi, so that is what will be used in this calculation. For this example, assume that we are at sea level, so ambient pressure is 14.7 psia.

We will need to subtract the 1 psi pressure loss from the ambient pressure to determine the Compressor Inlet Pressure (P_1).

$$P_{1c} = P_{amb} - \Delta P_{loss}$$

Where:

· P_{1c} = Compressor Inlet Pressure (psia)
· P_{amb} = Ambient Air pressure (psia)
· ΔP_{loss} = Pressure Loss due to Air Filter/Piping (psi)

P_{1c} = 14.7 - 1

= 13.7 psia

With this, we can calculate Pressure Ratio (Πc) using the equation.

$$\Pi c = {}^{P_{2c}}\!/_{P_{1c}}$$

For the 2.0 L engine:

$$\Pi c = {}^{43.1}\!/_{13.7}$$

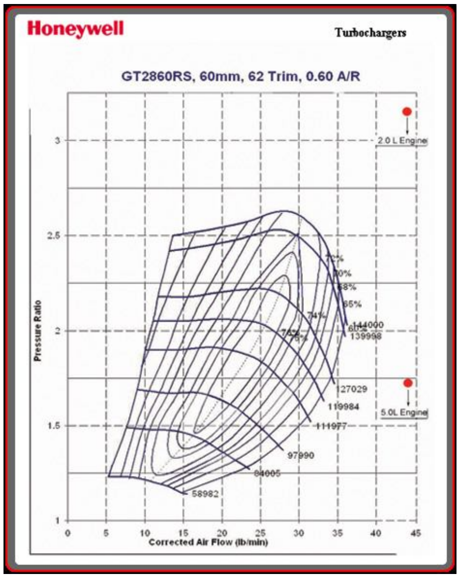
= 3.14

For the 5.0 L engine:

$$\Pi c = {}^{23.6}\!/_{13.7}$$

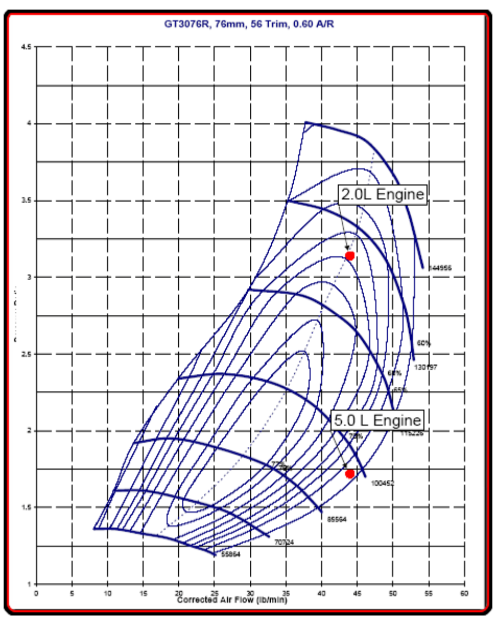
= 1.72

We now have enough information to plot these operating points on the compressor map. First we will try a GT2860RS. This turbo has a 60mm, 60 trim compressor wheel.



Clearly this compressor is too small, as both points are positioned far to the right and beyond the compressor's choke line.

Another potential candidate might be the GT3076R. This turbo has a 76mm, 56 trim compressor wheel:



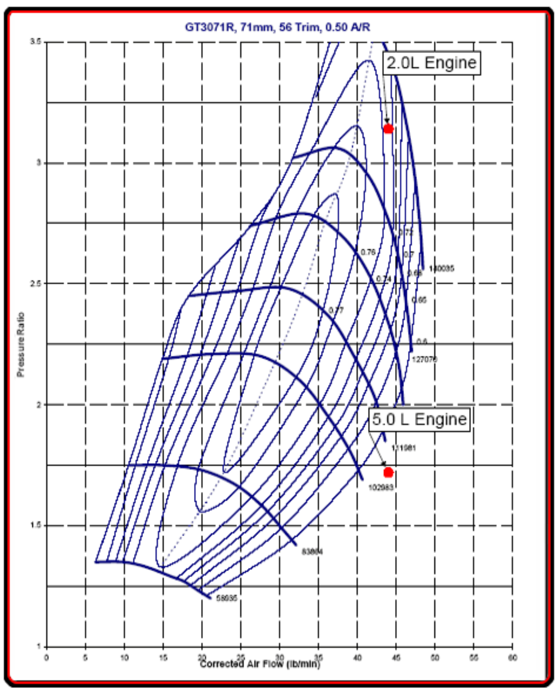
This is much better; at least both points are on the map! Let's look at each point in more detail.

For the 2.0L engine this point is in a very efficient area of the map, but since it is in the center of the map, there would be a concern that at a lower engine speeds that it would

be near or over the surge line. This might be ok for a high-rpm-biased powerband that might be used on a racing application, but a street application would be better served by a different compressor.

For the 5.0L engine, this looks like a very good street-biased powerband, with the lower engine speeds passing through the highest efficiency zone on the map, and plenty of margin to stay clear of surge. One area of concern would be turbo overspeed when revving the engine past peak power. A larger compressor would place the operating point nearer to the center of the map and would give some additional benefit to a high-rpm-biased powerband. We'll look at a larger compressor for the 5.0L after we figure out a good street match for the 2.0L engine.

So now lets look at a GT3071R, which uses a 71mm, 56 trim compressor wheel.



For the 2.0L engine, this is a much more mid-range-oriented compressor. The operating point is shifted a bit towards the choke side of the map and this provides additional surge margin. The lower engine speeds will now pass through the higher efficiency zones and give excellent performance and response.

For the 5.0L engine, the compressor is clearly too small and would not be considered.

Now that we have arrived at an acceptable compressor for the 2.0L engine, lets calculate a lower rpm point to put on the map to better get a feel for what the engine operating line will look like. We can calculate this using the following formula:

$$Wa = \frac{MAP * VE * N/_2 * Vd}{R * (460 + T_m)}$$

We'll choose the engine speed at which we would expect to see peak torque, based on experience or an educated guess. In this case we'll choose 5000rpm.

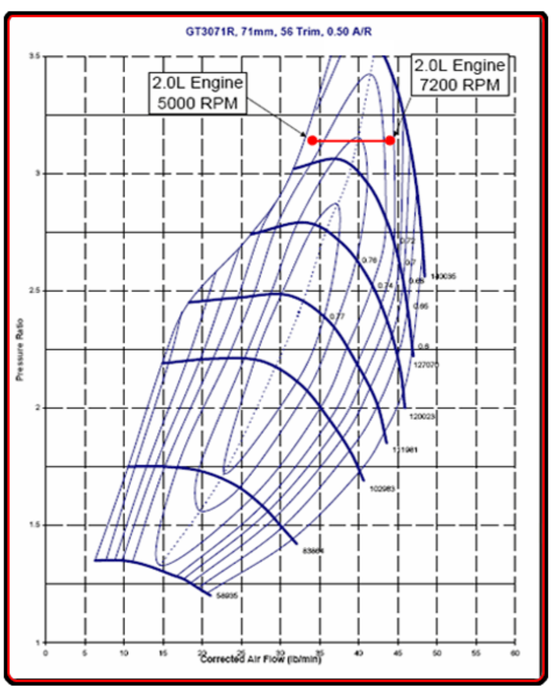
Where:

· Wa = $Airflow_{actual}$ (lb/min)
· MAP = Manifold Absolute Pressure (psia) =41.1 psia
· R = Gas Constant = 639.6
· Tm = Intake Manifold Temperature (degrees F) =130
· VE = Volumetric Efficiency = 0.98
· N = Engine speed (RPM) = 5000rpm
· Vd = engine displacement (Cubic Inches, convert from liters to CI by multiplying by 61, ex. 2.0 liters * 61 = 122 CI)

$$Wa = \frac{41.1 * 0.98 * 5000/_2 * 122}{639.6 * (460 + 130)}$$

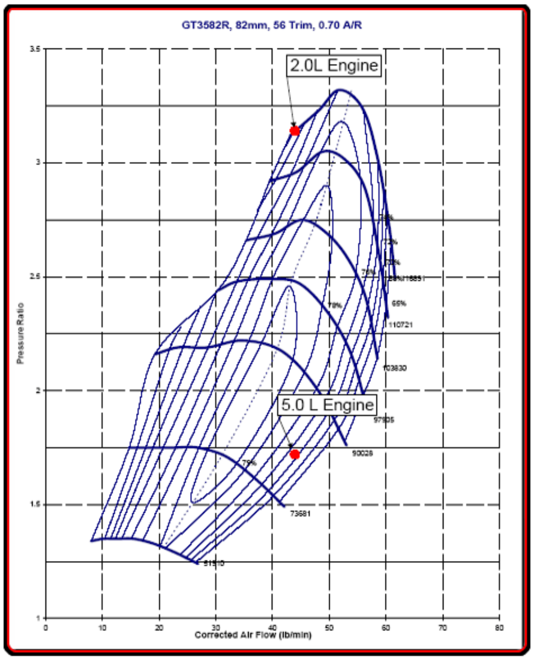
= 32.5 lb/min

Plotting this on the GT3071R compressor map gives the following operating points.



This gives a good representation of the operating line at that boost level, which is well suited to this map. At engine speeds lower than 5000rpm the boost pressure will be lower, and the pressure ratio would be lower, to keep the compressor out of surge.

Back to the 5.0 L engine. Let's look at a larger compressor's map. This time we will try a GT3582R with an 82mm, 56 trim compressor.



Here , compared to the GT3076R, we can see that this point is not quite so deep into choke and will give better high-rpm performance than the 76mm wheel. A further increase in wheel size would give even better high-rpm performance, but at the cost of low- and mid-range response and drivability.

Hopefully this has given a basic idea of what a compressor map displays and how to choose a compressor. As you can see, a few simple estimations and calculations can provide a good basis for compressor selection. If real data is available to be substituted in place of estimation, more accurate results can be generated.

TURBO DEVELOPMENT

Development

As turbochargers have to meet different requirements with regard to map height, map width, efficiency characteristics, moment of inertia of the rotor and conditions of use, new compressor and turbine types are continually being developed for various engine applications. Furthermore, different regional legal emission regulations lead to different technical solutions.

The compressor and turbine wheels have the greatest influence on the turbocharger's operational characteristics. These wheels are designed by means of computer programs which allow a three-dimensional calculation of the air and exhaust gas flows. The wheel strength is simultaneously optimized by means of the finite-element method (FEM), and durability calculated on the basis of realistic driving cycles.


CAD-assembled model of a turbocharger

Despite today's advanced computer technology and detailed calculation programs, it is testing which finally decides on the quality of the new aerodynamic components. The fine adjustment and checking of results is therefore carried out on turbocharger test stands.

Matching

The vital components of a turbocharger are the turbine and the compressor. Both are turbo-machines which, with the help of modeling laws, can be manufactured in various sizes with similar characteristics. Thus, by enlarging and reducing, the turbocharger range is established, allowing the optimal turbocharger frame size to be made available for various engine sizes. However, the transferability to other frame sizes is restricted, as not all characteristics can be scaled dimensionally. Furthermore, requirements vary in accordance with each engine size, so that it is not always possible to use the same wheel or housing geometries.

The model similarity and modular design principle, however, permit the development of turbochargers which are individually tailored to every engine. This starts with the selection of the appropriate compressor on the basis of the required boost pressure characteristic curve. Ideally, the full-load curve should be such that the compressor efficiency is at its maximum in the main operating range of the engine. The distance to the surge line should be sufficiently large.

The thermodynamic matching of the turbocharger is implemented by means of mass flow and energy balances. The air delivered by the compressor and the fuel fed to the engine constitute the turbine mass flow rate. In steady-state operation, the turbine and compressor power outputs are identical (free wheel condition). The matching calculation is iterative, based on compressor and turbine maps, as well as the most important engine data.

The matching calculation can be very precise when using computer programs for the calculated engine and turbocharger simulation. Such programs include mass, energy and material balances for all cylinders and the connected pipe work. The turbocharger enters into the calculation in the form of maps. Furthermore, such programs include a number of empirical equations to describe interrelationships which are difficult to express in an analytical way.

Testing

The turbocharger has to operate as reliably and for as long as the engine. Before a turbocharger is released for series production, it has to undergo a number of tests. This test program includes tests of individual turbocharger components, tests on the turbocharger test stand and a test on the engine. Some tests from this complex testing program are described below in detail.

Containment test
If a compressor or turbine wheel bursts, the remaining parts of the wheel must not penetrate the compressor or turbine housing. To achieve this, the shaft and turbine wheel assembly is accelerated to such a high speed that the respective wheel bursts. After bursting, the housing's containment safety is assessed. The burst speed is typically 50 percnt; above the maximum permissible speed.

Low-Cycle Fatigue Test (LCF test)
The LCF test is a load test of the compressor or turbine wheel resulting in the component's destruction. It is used to determine the wheel material load limits. The compressor or turbine wheel is installed on an overspeed test stand. The wheel is accelerated by means of an electric motor until the specified tip speed is reached and then slowed down. On the basis of the results and the component's S/N curve, the expected lifetime can be calculated for every load cycle.

Rotor dynamic measurement
The rotational movement of the rotor is affected by the pulsating gas forces on the

turbine. Through its own residual imbalance and through the mechanical vibrations of the engine, it is stimulated to vibrate. Large amplitudes may therefore occur within the bearing clearance and lead to instabilities, especially when the lubricating oil pressures are too low and the oil temperatures too high. At worst, this will result in metallic contact and abnormal mechanical wear.

The motion of the rotor is measured and recorded by contactless transducers located in the suction area of the compressor by means of the eddy current method. In all conditions and at all operating points, the rotor amplitudes should not exceed 80 % of maximum possible values. The motion of the rotor must not show any instability.

Start-stop test

The temperature drop in the turbocharger between the gases at the hot turbine side and at the cold compressor inlet can amount to as much as 1000 °C in a distance of only a few centimeters. During the engine's operation, the lubricating oil passing through the bearing cools the center housing so that no critical component temperatures occur. After the engine has been shut down, especially from high loads, heat can accumulate in the center housing, resulting in coking of the lubricating oil. It is therefore of vital importance to determine the maximum component temperatures at the critical points, to avoid the formation of lacquer and carbonized oil in the turbine-side bearing area and on the piston ring.

After the engine has been shut down at the full-load operating point, the turbocharger's heat build-up is measured. After a specified number of cycles, the turbocharger components are inspected. Only when the maximum permissible component temperatures are not exceeded and the carbonized oil quantities around the bearing are found to be low, is this test considered passed.

Cyclic endurance test

During engine operation, the waste gate is exposed to high thermal and mechanical loads. During the waste gate test, these loads are simulated on the test stand

The checking of all components and the determination of the rates of wear are included in the cycle test. In this test, the turbocharger is run on the engine for several hundred hours at varying load points. The rates of wear are determined by detailed measurements of the individual components, before and after the test.

TURBO CONTROL SYSTEMS

The drivability of passenger car turbo engines must meet the same high requirements as naturally aspirated engines of the same power output. That means, full boost pressure must be available at low engine speeds. This can only be achieved with a boost pressure control system on the turbine side.

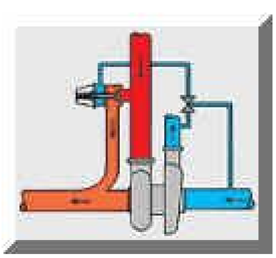
Control by turbine-side bypass (wastegate)

The turbine-side bypass is the simplest form of boost pressure control. The turbine size is chosen such that torque characteristic requirements at low engine speeds can be met and good vehicle drivability achieved. With this design, more exhaust gas than required to produce the necessary boost pressure is supplied to the turbine shortly before the maximum torque is reached. Therefore, once a specific boost pressure is achieved, part of the exhaust gas flow is fed around the turbine via a bypass. The wastegate which opens or closes the bypass is usually operated by a spring-loaded diaphragm in response to the boost pressure.

Boost Controller

Today, electronic boost pressure control systems are increasingly used in modern passenger car diesel and petrol engines. When compared with purely pneumatic control, which can only function as a full-load pressure limiter, a flexible boost pressure control allows an optimal part-load boost pressure setting. This operates in accordance with various parameters such as charge air temperature, degree of timing advance and fuel quality. The operation of the flap corresponds to that of the previously described actuator. The actuator diaphragm is subjected to a modulated control pressure instead of full boost pressure.

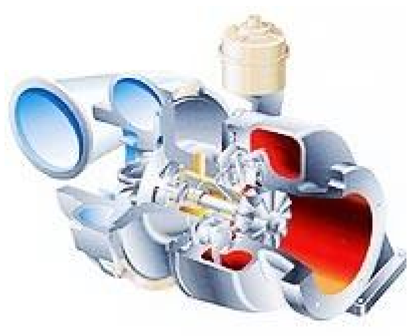
Boost pressure control of a turbocharged petrol engine by proportional control Pressure



This control pressure is lower than the boost pressure and generated by a proportional valve. This ensures that the diaphragm is subjected to the boost pressure and the pressure at the compressor inlet in varying proportions. The proportional valve is controlled by the engine electronics. For diesel engines, a vacuum-regulated actuator is used for electronic boost pressure control.

Variable turbine geometry

The variable turbine geometry allows the turbine flow cross-section to be varied in accordance with the engine operating point. This allows the entire exhaust gas energy to be utilized and the turbine flow cross-section to be set optimally for each operating point. As a result, the efficiency of the turbocharger and hence that of the engine is higher than that achieved with the bypass control.

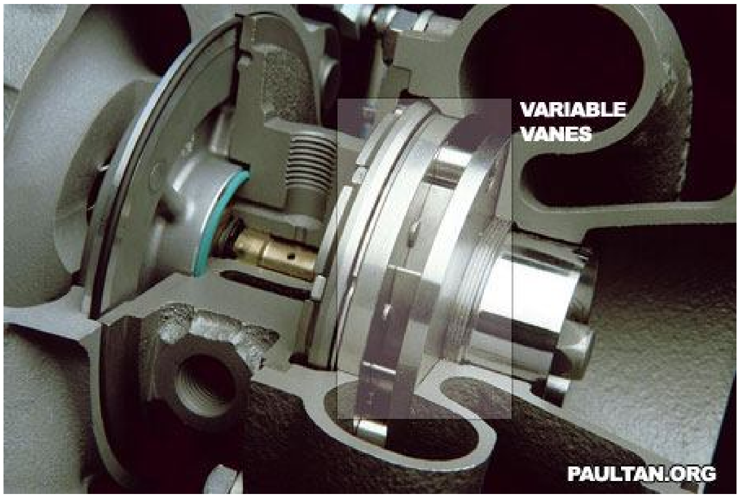


Turbocharger for truck applications with variable turbine geometry (VTG)

How does Variable Turbine Geometry work?

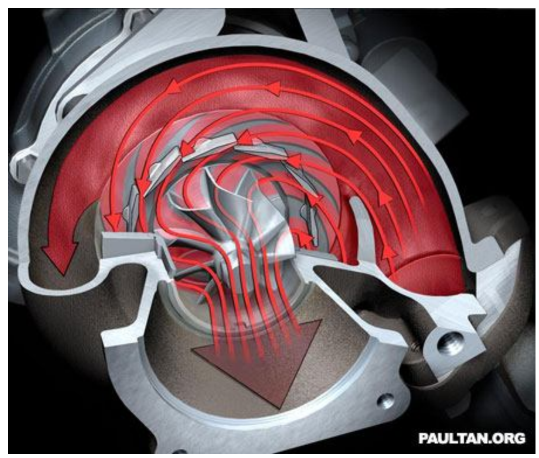
Variable Turbine Geometry technology is the next generation in turbocharger technology where the turbo uses variable vanes to control exhaust flow against the turbine blades. See, the problem with the turbocharger that we've all come to know and love is that big turbos do not work well at slow engine speeds, while small turbos are fast to spool but run out of steam pretty quick. So how doVTG turbos solve this problem?

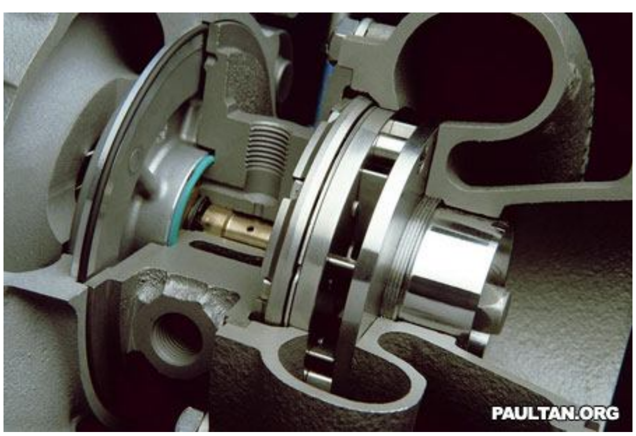
A Variable Turbine Geometry turbocharger is also known as a variable geometry turbocharger (VGT), or aVariable Nozzle Turbine (VNT). A turbocharger equipped with Variable Turbine Geometry has little movable vanes which can direct exhaust flow onto the turbine blades. The vane angles are adjusted via an actuator. The angle of the vanes vary throughout the engine RPM range to optimize turbine behaviour.



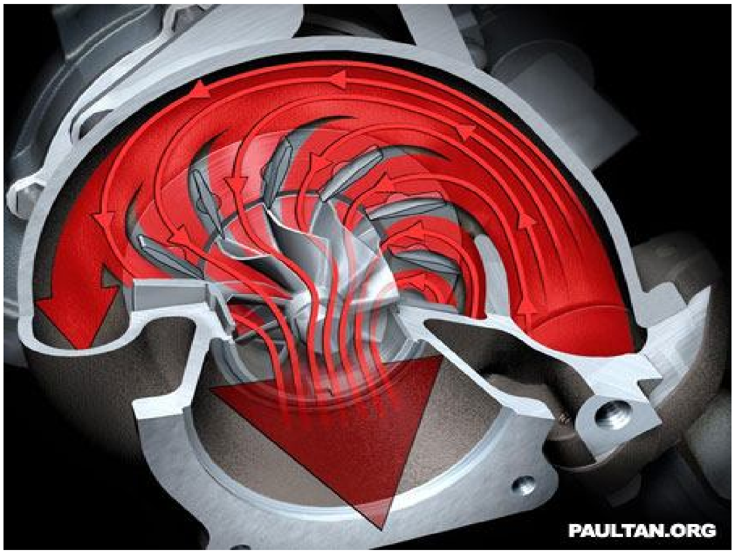
In the 3D illustration above, you can see the vanes in a angle which is almost closed. I have highlighted the variable vanes so you know which is which. This position is optimized for low engine RPM speeds, pre-boost.

In this cut-through diagram, you can see the direction of exhaust flow when the variable vanes are in an almost closed angle. The narrow passage of which the exhaust gas has to flow through accelerates the exhaust gas towards the turbine blades, making them spin faster. The angle of the vanes also directs the gas to hit the blades at the proper angle.





Above are how the VGT vanes look like when they are open. I've not highlighted where the vanes are in this image since you already know where they are, as to not spoil the mechanical beauty that it is 😛



This cut-through diagram shows the exhaust gas flow when the variable turbine vanes are fully open. The high exhaust flow at high engine speeds are fully directed onto the turbine blades by the variable vanes.

Variable Turbine Geometry has been used extensively in turbodiesel engines since the 1990s, but it has never been on a production petrol turbocharged car before until the new. This is because petrol engine exhaust gases are alot hotter than diesel engine exhaust gas, so generally the material used to make VTG turbos could not stand this heat. The uses a BorgWarner VTG turbocharger which uses special materials derived from aerospace technology, hence solving the temperature problem.

Automotive applications

Turbocharging is very common on diesel engines in conventional automobiles, in trucks, locomotives, for marine and heavy machinery applications. In fact, for current automotive applications, non-turbocharged diesel engines are becoming increasingly rare[*citation needed*]. Diesels are particularly suitable for turbocharging for several reasons:

- Naturally-aspirated diesels develop less power than gasoline engines of the same displacement, and will weigh significantly more because diesel engines require heavier, stronger components. This gives such engines a poor power-to-weight ratio, which turbocharging can dramatically improve with only slight additional weight.
- Diesel engines operate within a speed range, facilitating the use of a narrowly-optimized turbocharger.
- Diesel engines are not prone to the detonation that arises from high (or forced) cylinder pressure and can damage gasoline engines.
- Unlike gasoline (petrol) engines which experience higher fuel consumption when turbocharged, turbocharging can reduce the fuel consumption of a diesel engine.

The turbocharger's small size and low weight have production and marketing advantage to vehicle manufacturers. By providing naturally-aspirated and turbocharged versions of one engine, the manufacturer can offer two different power outputs with only a fraction of the development and production costs of designing and installing a different engine. The compact nature of a turbocharger mean that bodywork and engine compartment layout changes to accommodate the more powerful engine are not needed or minimal. Parts commonality between the two versions of the same engine reduces production and servicing costs.

Today, turbochargers are most commonly used on gasoline engines in high-performance automobiles and diesel engines in transportation and other industrial equipment[*citation needed*]. Small cars in particular benefit from this technology, as there is often little room to fit a large engine. Volvo and Saab have produced turbocharged cars for many years, the turbo Porsche 944's acceleration performance was very similar to that of the larger-engined non-turbo Porsche 928, and Chrysler Corporation built numerous turbocharged cars in the 1980s and 1990s.

Aircraft

A more natural use of the turbocharger is with aircraft engines. As an aircraft climbs to higher altitudes the pressure of the surrounding air quickly falls off. At 5,486 m (18,000 ft) the air is at half the pressure of sea level, and the airframe only experiences half the aerodynamic drag. However, since the charge in the cylinders is being pushed in by this air pressure, it means that the engine will normally produce only half-power at full throttle at this altitude. Pilots would like to take advantage of the low drag at high altitudes in order to go faster, but a naturally aspirated engine will not produce enough power at the same altitude to do so.

Altitude effects

A turbocharger remedies this problem by compressing the air back to sea-level pressures; or even much higher; in order to produce rated power at high altitude. Since the size of the turbocharger is chosen to produce a given amount of pressure at high altitude, the turbocharger is over-sized for low altitude. The speed of the turbocharger is controlled by a wastegate. Early systems used a fixed wastegate, resulting in a turbocharger that functioned much like a supercharger. Later systems utilized an adjustable wastegate, controlled either manually by the pilot or by an automatic hydraulic or electric system. When the aircraft is at low altitude the wastegate is usually fully open, venting all the exhaust gases overboard. As the aircraft climbs and the air density drops, the wastegate must continually close in small increments to maintain full power. The altitude at which the wastegate is full closed and the engine is still producing full rated power is known as the *critical altitude.*

Temperature considerations

One disadvantage of turbocharging is that compressing the air increases its temperature, which is true for any method of forced induction. This causes multiple problems. Increased temperatures can lead to detonation and excessive cylinder head temperatures. In addition, hotter air is less dense, so fewer air molecules enter the cylinders on each intake stroke, resulting in an effective drop in volumetric efficiency which works against the efforts of the turbocharger to increase volumetric efficiency.

Aircraft engines generally cope with this problem in one of several ways. The most common one is to add an intercooler or aftercooler somewhere in the air stream between the compressor outlet of the turbocharger and the engine intake manifold. Intercoolers and aftercoolers are types of heat exchangers which cause the compressed air to give up some of its heat energy to the ambient air. In the past, some aircraft featured anti-detonant injection for takeoff and climb phases of flight, which performs the function of cooling the fuel/air charge before it reaches the cylinders.

In contrast, modern turbocharged aircraft usually forego any kind of temperature compensation, because the turbochargers are generally small and the manifold pressures created by the turbocharger are not very high. Thus the added weight, cost, and complexity of a charge cooling system are considered to be unnecessary penalties. In those cases the turbocharger is limited by the temperature at the compressor outlet, and the turbocharger and its controls are designed to prevent a large enough temperature rise to cause detonation. Even so, in many cases the engines are designed to run rich in order to use the evaporating fuel for charge cooling.

Advantages and disadvantages

Advantages

- More specific power over naturally aspirated engine. This means a turbocharged engine can achieve more power from same engine volume.
- Better thermal efficiency over both naturally aspirated and supercharged engine when under full load (*i.e.* on boost). This is because the excess exhaust heat and pressure, which would normally be wasted, contributes some of the work required to compress the air.
- Weight/Packaging. Smaller and lighter than alternative forced induction systems and may be more easily fitted in an engine bay.
- Fuel Economy. Although adding a turbocharger itself does not save fuel, it will allow a vehicle to use a smaller engine while achieving power levels of a much larger engine, while attaining near normal fuel economy while off boost/cruising. This is because without boost, less fuel is used to create a proper air/fuel ratio.

Disadvantages

- Lack of responsiveness if an incorrectly sized turbocharger is used. If a turbocharger that is too large is used it reduces throttle response as it builds up boost slowly otherwise known as "lag". However, doing this may result in more *peak* power.
- Boost threshold. A turbocharger starts producing boost only above a certain rpm due to a lack of exhaust gas volume to overcome inertia of rest of the turbo propeller. This results in a rapid and nonlinear rise in torque, and will reduce the usable power band of the engine. The sudden surge of power could overwhelm the tires and result in loss of grip, which could lead to understeer/oversteer, depending on the drivetrain and suspension setup of the vehicle. Lag can be disadvantageous in racing, if throttle is applied in a turn, power may unexpectedly increase when the turbo spools up, which can cause excessive wheelspin.
- Cost. Turbocharger parts are costly to add to naturally aspirated engines. Heavily modifying OEM turbocharger systems also require extensive upgrades that in most cases requires most (if not all) of the original components to be replaced.
- Complexity. Further to cost, turbochargers require numerous additional systems if they are not to damage an engine. Even an engine under only light boost requires a system for properly routing (and sometimes cooling) the lubricating oil, turbo-specific exhaust manifold, application specific downpipe, boost regulation. In addition inter-cooled turbo engines require additional plumbing, while highly tuned turbocharged engines will require extensive upgrades to their lubrication, cooling, and breathing systems; while reinforcing internal engine and transmission parts.

TURBO RECOMMENDATIONS

What is good for a turbocharger?

The turbocharger is designed such that it will usually last as long as the engine. It does not require any special maintenance; and inspection is limited to a few periodic checks.

To ensure that the turbocharger's lifetime corresponds to that of the engine, the following engine manufacturer's service instructions must be strictly observed:

- Oil change intervals
- Oil filter system maintenance
- Oil pressure control
- Air filter system maintenance

What is bad for a turbocharger?

90% of all turbocharger failures are due to the following causes:

- Penetration of foreign bodies into the turbine or the compressor
- Dirt in the oil
- Inadequate oil supply (oil pressure/filter system)
- High exhaust gas temperatures (ignition system/injection system)
- These failures can be avoided by regular maintenance. When maintaining the air filter system, for example, care should be taken that no trash or foreign material gets into the turbocharger.

Failure diagnosis

If the engine does not operate properly, one should not assume that the turbocharger is the cause of failure. It often happens that fully functioning turbochargers are replaced even though the failure does not lie here, but with the engine.

Only after all these points have been checked should one check the turbocharger for faults. Since the turbocharger components are manufactured on high-precision machines to close tolerances and the wheels rotate up to 300,000 rpm, turbochargers should be inspected by qualified specialists only.

TURBO TROUBLESHOOTING

All too frequently, serviceable turbochargers are removed from engines before the cause of the problem has been determined. Always inspect and assess turbocharger condition before removal from the engine.

Should removal of the turbo become necessary, try to determine if the connections were tight and without leaks while you are removing the hoses, clamps or connections. Once disassembly has been completed, it may be difficult or impossible to substantiate the conditions that caused the problem.

Problems experienced in the field can most often be corrected by system troubleshooting. Immediate or early failure of a replacement turbocharger may be related to:

- The incomplete correction of the problem that caused the need for the replacement.
- Problems introduced during the replacement.
- A defective turbocharger.

A turbocharger that has operated successfully is very unlikely to be found defective at a later date. Speed and temperature normally seen in turbocharger operation usually identify defects very quickly. Installation or engine system problems can also show up immediately upon replacement. Don't be too quick to blame the turbo for operational problems if the turbocharger spins freely and has not rubbed the housing.

It should be emphasized that a turbocharger does not basically change the operating characteristics of an engine. A turbocharger is not a power source within itself. The turbo's only function is to supply a greater volume of compressed air to the engine so that more fuel can be burned to produce more power. It can function only as dictated by the flow, pressure and temperature in the engine exhaust gas.

Turbochargers are an integral component of a complete operating system. Only by convenience is a turbo external or 'bolt-on" in installation. It is no less dedicated than an engine's camshaft or pistons. Understanding how a turbocharger is part of a complete engine management system is essential in successfully diagnosing and repairing problems. Likewise, a better understanding of some of a turbocharger's features can be helpful when determining that a turbo is damaged or defective, and installing it correctly the first time, every time.

Verify that the turbocharger is the correct configuration for the application. Assembly and component part numbers may both need attention. This is particularly important because the "matching" process requires subtle component differences. Part number

checks are necessary because some of the possible discrepancies will not be apparent to the untrained eye.

A turbocharger cannot correct or overcome such things as malfunctions or deficiencies in the engine fuel system, timing, plugged air cleaners, faulty liners, etc. Therefore, if a turbocharged engine system has malfunctioned and the turbocharger has been examined and determined to be operational, proceed with troubleshooting, as though the engine were non-turbocharged. Simply replacing a good turbocharger with another will not correct basic engine deficiencies.

Common Symptoms

Turbochargers and engines have common problem symptoms.

- Engine Lacks Power
- Engine Exhaust Smoke
- Oil Consumption
- Noisy Operation

Any of these symptoms could be the result of an internal engine problem and might not involve the turbocharger at all. The following on-engine troubleshooting guide is designed to quickly determine turbocharger condition and prevent unnecessary removal.

On Engine Troubleshooting

Many of the problem causing conditions will appear in direct or inverse proportion to the power output. For example, there may be a problem at idle that is unnoticeable at full power or visa versa. The following procedures are an overall evaluation involving varying operational conditions. On-engine troubleshooting will also help to expose any external or engine related causes of turbocharger failure that must be corrected to prevent the failure of a replacement unit.

The most efficient way to troubleshoot a performance complaint is to proceed through all of the steps in the order presented before making a final determination of the service required. It is extremely important that all in-service problem areas are examined before any single one is corrected.

In some instances, corrective service may lead you to turbocharger Damage Analysis and, depending on the results of these inspections and/or the turbocharger model, you may also have to measure the bearing clearances or test the wastegate device. Those inspections, as well as a detailed analysis of problems that may be exposed here, are also covered in turbocharger Damage Analysis.

On-engine troubleshooting consists of several basic steps that should be taken before the turbocharger is removed from the engine. Any external or engine-related faults found must be corrected before a replacement turbocharger is installed.

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running, make sure it is cool before beginning.

Warning: Operating the turbocharger without the inlet duct and air filter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

The basic steps to troubleshooting are as follows:

VISUAL AND MECHANICAL CHECKS

Inspect the turbocharger exterior and installation. Listen for unusual mechanical noises. Visually check and test for leaks, blockage, high heat, restrictions or conditions that have allowed wheels to contact the housings. Leaks that are seemingly small and insignificant at idle or low power can greatly affect air/fuel ratios and pressures within the housing and full power. At full power, those leaks become problematic.

- Listen for unusual mechanical noise and watch for vibration.
- Listen for a high pitched noise. It can indicate air or gas leaks.
- Listen for noise level cycling. It can indicate a restriction in the air cleaner or ducting.
- Inspect for missing or loose nuts, bolts, clamps and washers.
- Inspect for loose or damaged intake and exhaust manifolds and their ducting and clamps.
- Inspect for damaged or restricted oil supply and drain lines.
- Inspect for cracked or deteriorating turbocharger housings.
- Inspect for external oil or coolant leakage; external deposits (indicates air, oil, exhaust or coolant leakage).
- Inspect for obvious heat discoloration.
- Inspect for obviously restricted air filter.
- Check the wastegate for free movement and damage. Be sure that hoses are in good condition and that the connections are tight. Check the calibration and control system according to the original equipment specifications.
- Verify that the turbocharger is the correct configuration for the application.

Remember, correcting these problems does not in it self remove any residues that were the indicators of the problem. The remaining residues often cause inaccurate turbocharger evaluation. Incorrect turbocharger evaluation may result after the situation

has been corrected and the residues remain. For example, an air filter replaced just previous to your inspection would lead you to conclude that air blockage is not the problem even though the residue indicates blockage.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, then the unit should be replaced at this time and corrective actions taken to prevent reoccurrence.

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running make sure it is cool before beginning.

Warning: Operating the turbocharger without the inlet duct and air filter connected can result In personal injury. Equipment damage may result from foreign objects entering the turbocharger.

TURBINE WHEEL AND TURBINE HOUSING CHECKS

Remove the ducting from the turbine outlet. Using an inspection light:

Inspect the turbine for evidence of foreign object damage. This is usually not easily visible from the turbine outlet unless the damage is severe. Determine the source of the object and check for possible engine damage. Figure 20 highlights where turbine wheel rub frequently occurs.

Turn the rotating assembly by hand and feel for dragging or binding; also check by pushing the assembly sideways while turning. The wheel should turn freely and without any rubbing or scraping noises. If there are obvious signs of wheel rub or that the turbine housing has been operated at excessive temperatures, then the turbocharger is damaged and must be replaced. If you are still not sure whether the wheel is rubbing, inspect the bearing clearances after completing this section.

Look for evidence of oil leakage. If oil deposits are found, then determine whether the oil is from the engine or from the turbocharger center housing. Some oil residues may be cleaned; heavy oil residues may require replacement. If the oil is from the center housing, then remove the oil drain line and look into the turbocharger drain opening and drain line with an inspection light. Check for an oily, sludge build-up on the shaft between the bearing journals, in the drain cavity, and in the drain line.

Check the following to determine the cause of the problem and effect corrections as necessary:

- Restricted draining or high crankcase pressure can raise the pressure of the center housing drain area above the pressure in the turbine housing forcing oil in that direction.
- PCV flow control valves on spark ignition engines must operate as one way check valves when boost is developed. This reverses the direction of flow in the ventilation system. A partially closed PCV allows manifold boost to pressurize the crankcase.
- Damaged oil drain line.
- Improper line routing (more than 35 degrees from vertical or any sharp bends) or routings close to exhaust manifolds.
- Submerged drain line from too high an oil level or equipment operated at extreme angle.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, the unit should be replaced at this time and corrective actions taken to prevent reoccurrence.

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running make sure it is cool before beginning.

Warning: Operating the turbocharger without the inlet duct and air filter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

COMPRESSOR WHEEL AND COMPRESSOR HOUSING CHECKS

Remove the ducting from the compressor inlet. Using an inspection light:

Inspect the compressor for evidence of foreign object damage. If the wheel is damaged, the foreign object probably entered through the intake system. Remember that the origin of foreign object damage should be identified. Foreign objects usually come from human error or deteriorated intake systems. Determine the source of the object, clean the system, and check for possible engine damage.

Turn the rotating assembly by hand and feel for dragging or binding; also check by pushing the assembly sideways while turning. Look for any evidence of wheel rub. Wheel rub can be caused by loose, distorted, or binding housings as well as damaged bearings. If there is still any doubt, inspect the bearing clearances.

Look for evidence of oil leakage. The compressor side is most sensitive to a restricted air inlet. Oil in the compressor outlet does not prove turbocharger seal leakage. Oil from crankcase ventilation or other oil sources can be confused with compressor-side oil leaks. The turbocharger compressor can take oil vapor and expel it as liquid oil.

General engine condition greatly affects engine crankcase ventilation system operation. Follow manufacturer's recommendations. Other factors that can cause oil leakage into the compressor are detailed in the Troubleshooting charts. Compressor oil leaks can result in oil accumulations in the charge-air cooler. When all problems have been corrected this oil can be transferred into the engine. I f oil accumulation occurs, it will require draining and cleaning of the charge-air cooler.

Correct any installation problems after completing the rest of this procedure. If turbocharger parts are damaged, the unit should be replaced at this time and

Refer to the engine manufacturer's service instructions for inspection requirements and replacement specifications.

CAUTION: Do not run the engine during these procedures. If the engine has been running, make sure it is cool before beginning.

Warning: Operating the turbocharger without the inlet duct and air filter connected can result in personal injury. Equipment damage may result from foreign objects entering the turbocharger.

ROTATING ASSEMBLY CHECK

Check for signs of a sludged or coked center housing. A sludged or coked center housing will not likely be evident by inspecting the end housings. Evidence of this condition will be found in advanced cases by looking for oil deposits in the oil inlet. Also check the oil drain area.

Turn the rotating assembly by hand and feel for dragging or binding; also check by pushing the assembly sideways while turning. Look for any evidence of wheel rub. Wheel rub can be caused by loose, distorted, or binding housings as well as damaged bearings. If there is still any doubt, inspect the bearing clearances.

Look for evidence of leakage; either oil and/or coolant.

- Loose or improper connections.
- Improper gaskets or gasket material.
- Casting porosity.
- Improperly drilled holes.

CHECK RADIAL AND AXIAL BEARING CLEARANCES

If none of the previous steps have revealed any turbocharger faults, or if the evidence is not conclusive, this procedure will show if the unit is worn or damaged internally to the point of needing replacement.

Radial Journal Bearing Clearance Note: Due to the unique location of the internal opening in the center housing casting on models T45 and T51, access to the shaft wheel at this point is difficult.

Check the radial clearance of the journal bearings as follows:

For all models, except T45 and T51, attach the turbocharger gauge set to the unit so that the dial indicator plunger extends through the oil drain port and contacts the shaft of the turbine wheel assembly.

For models T45 and T51 only, place the special curved end of the gauge arm in contact with the wheel shaft through the oil outlet port and the internal opening in the casting.

- The dial indicator shaft is then placed in contact with the exposed portion of the gauge arm at a point equidistant from the gauge arm pivot and a point of contact with the wheel shaft, with the arm kept in contact with the shaft by the spring action of the dial indicator plunger.
- Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go away from the dial indicator plunger.
- Set the dial indicator to zero.
- Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go toward the dial indicator plunger. Note the maximum shaft movement shown on the indicator dial.
- To make sure that the dial indicator reading is the maximum possible, roll the wheels slightly in one direction and then the other while applying pressure.
- Manually apply pressure equally and simultaneously to the compressor and turbine wheels to move the shaft as far as it will go away from the dial indicator plunger. Make sure the dial indicator pointer returns to zero.
- Repeat steps "b" through "f" several times to make sure that the maximum bearing radial clearance, as indicated by the maximum shaft movement, has been measured.
- Compare the maximum clearance measured to the specification for bearing radial clearance for the model turbocharger being tested, as found in the specifications section of your catalog. If the measurement is within the specification, the journal bearings are in good condition. If the measurement is not within the specification, the turbocharger is worn or damaged internally and must be replaced.

Axial (Thrust) Bearing Clearance Check the axial clearance of the thrust bearing as follows:

- Clean the hub end of the turbine wheel assembly.

- Attach a turbocharger gage set to the turbine end of the turbocharger so that the dial indicator plunger rests on the hub end of the turbine wheel assembly.
- Manually apply pressure to the compressor wheel and turbine wheel assembly to move the assembly as far as it will go away from the turbine end of the turbocharger (away from the dial indicator plunger).
- Set the dial indicator to zero.
- Manually apply pressure to the compressor wheel and turbine wheel assembly to move the assembly as far as it will go toward the turbine end of the turbocharger (toward the dial indicator plunger). Note the maximum shaft movement shown on the indicator dial.
- Repeat steps "c" through "e" several times to make sure that the maximum bearing axial clearance, as indicated by the maximum turbine wheel assembly movement, has been measured.
- Compare the minimum and maximum clearance measured to the specification for bearing axial clearance for the model turbocharger being tested, as found in the specifications section of your catalog.

If the measurement is within the specification, the thrust bearing is in good condition. If no other faults have been found in previous steps, the turbocharger is likely not at fault in the complaint. Troubleshoot the engine as instructed in the engine manufacturer's service manual.

If the turbocharger was recently replaced or overhauled, make certain that the proper unit was installed or that the right parts were used in the overhaul. A turbocharger can appear to be right for the installation, but if the turbine and compressor components are not identical to those recommended by the engine manufacturer, performance and service life can suffer.

If the measurement is out of specification, the turbocharger is worn or damaged internally and must be replaced.

WASTEGATE ASSEMBLY CHECK

Wastegates may be an integral part of the turbine housing or a separate device plumbed into the exhaust system. Actuators are connected directly to the compressor outlet or work in conjunction with the engine management system. Engine manufacturers supply specific information on wastegates because of their arrangement within the engine management system.

Actuators spring pre-load may be high enough to make checking for free movement by hand difficult or impossible. Visually check for obstacles that can prevent movement or closure. Inverted exhaust pipe connection studs can prevent some wastegates from opening. Stress relief cracking may be found around the relief port in turbine housings. Cracks that do not extend beyond the wastegate valve do not present a problem.

CAUTION! When checking an actuator do not over pressurize because the diaphragm may become damaged. Swing valve actuators should move smoothly and show no decay when subjected to calibration pressures. Many poppet valve units have a hollow stem that opens in the guide giving a small leak when pressurized. Poppet valve units also depend to some degree on engine vibration to overcome static friction. When checking this type unit, light tapping of the housing will usually provide an accurate calibration check.

Original equipment specifications for calibration should be closely followed because they are established to interact with the entire engine management system. The calibration pressure is not necessarily a reflection of expected manifold boost because other pressures act on the valve. In many cases pressure to the actuator is overridden by the engine control system to vary the amount of boost depending upon conditions.

Many actuators are mounted on a bracket away from high temperatures. Problems may come from bending of the brackets or rods. High temperatures from exhaust leaks, corrosion, or other loose or damaged components can also result in wastegate problems.

When all else fails ask somebody that knows.

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